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DEVELOPMENT OF A HIGH ENERGY STORAGE FLYWHEEL MODULE.(U)
MAY 78 D R HODSON DAAG53-75-C-0278
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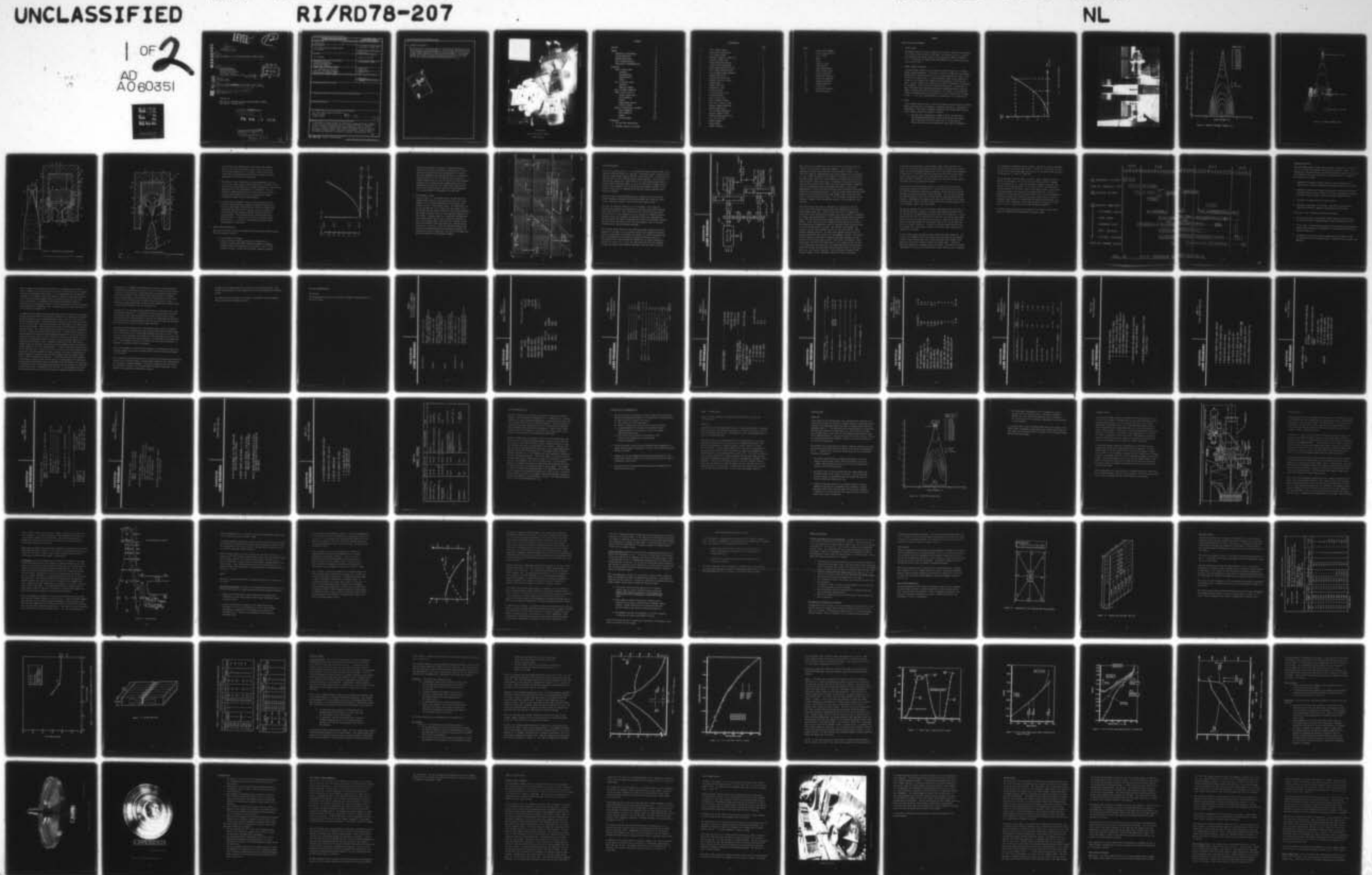

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(6) DEVELOPMENT OF A HIGH ENERGY STORAGE FLYWHEEL MODULE.

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A three-phase 35 month program was conducted to design and build a complete flywheel energy storage system rated at 30 KWH (40 HP-HRS) at 15,000 RPM. In the first phase, tradeoff studies were completed to select optimum materials, flywheel configuration, interface characteristics and to define performance characteristics. During the second phase, a detail design of the system was generated, including all drawings, specifications, and

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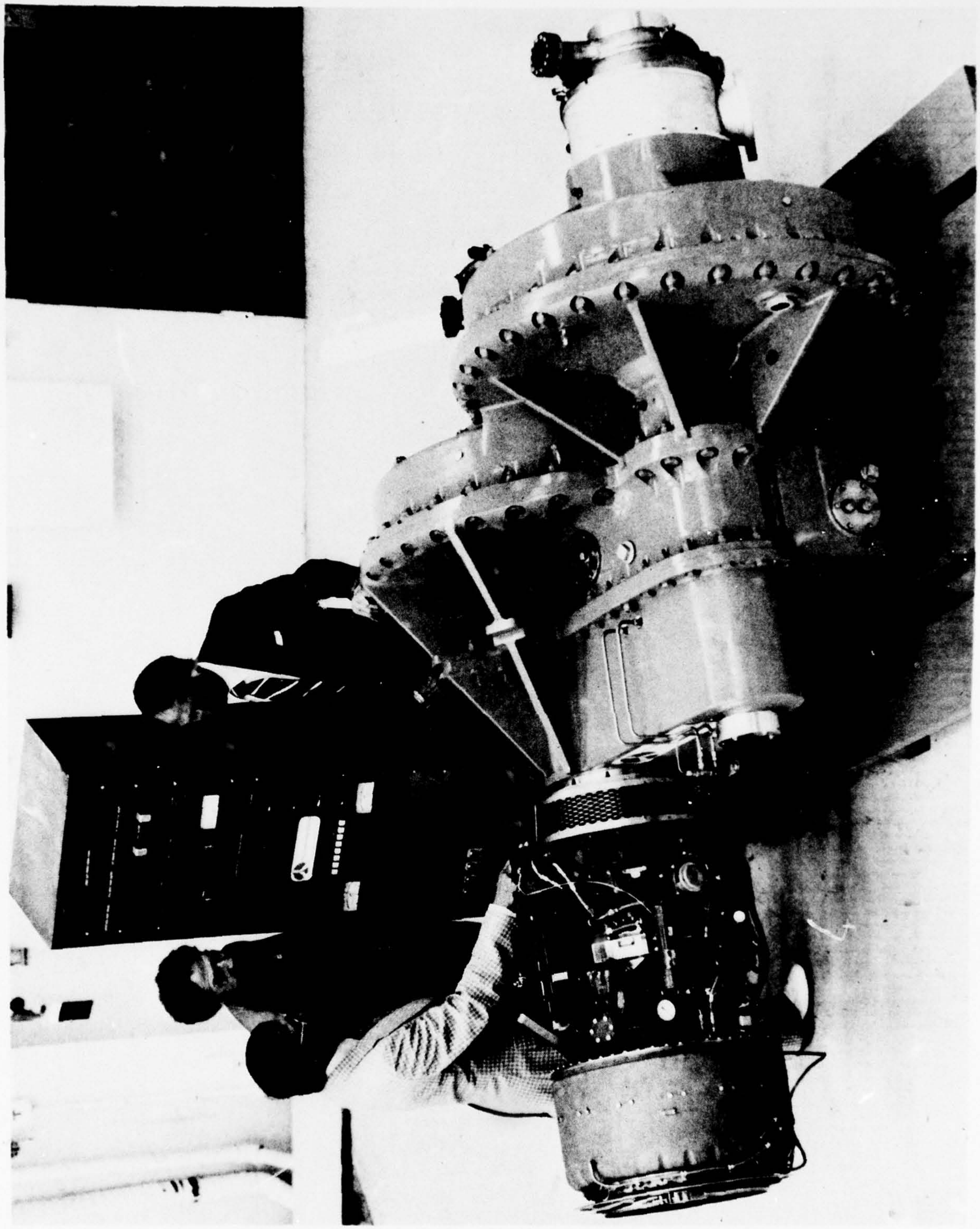
20. ABSTRACT (Continued)

descriptive data. A subscale model of the fullscale flywheel was also fabricated and tested during phase 2. The third and final phase of the current program consisted of fabrication and assembly of the fullscale 30 KWH flywheel module and associated control equipment. All program hardware was stored at Rocketdyne pending decisions to proceed with test of the system identified as Rocketdyne model RS-31.



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Frontisphoto

RS-31 High Energy Storage

Flywheel Module

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SUMMARY

OBJECTIVES AND ACHIEVEMENTS

1. STORED ENERGY

One primary objective of this program was to produce a flywheel system capable of storing at least 30 KWH at 15,000 RPM. This objective has been achieved and surpassed with rotors capable of storing almost 32 KWH at 15,000 RPM. Fig. 1 describes the energy-speed relationship for the system using rotors as pictured by Fig. 2.

2. ENERGY STORAGE EFFICIENCY

A second primary objective was to advance the state of the art in flywheel energy storage efficiency. To this end, a 2400 lb 30 KWH rotor was proposed wherein the 12.5 WH/LB rating was to be achieved operating at a stress level of 185 KSI as shown in Fig 3 A more meaningful index is given however, by rating at the ultimate stress capability of the material (220 KSI in this case). By this standard, the proposed rotor (Ref. 1) would have an efficiency of 14.9 WH/LB. In the interest of safety and efficiency, a unique new flywheel configuration was developed during this program wherein the rotor assembly's ultimate efficiency was increased to 17.1 WH/LB. This achievement allowed the working stress to be reduced to 140 KSI at rated speed (as shown in Fig. 4) yet total stored energy was increased by 6.7% as noted above.

3. SAFETY

A third primary objective was to achieve the highest practical degree of product safety consistent with the preceding performance objectives. Significant advancements in the field of flywheel technology were accomplished under the safety catagory during this program as follows:

a) The new rotor configuration as shown in Fig. 5 was conceived and demonstrated wherein the ultimate mode of failure releases only a small part of the disc mass Fig. 6 in the event of failure (versus total disc breakup for other types of flywheels).

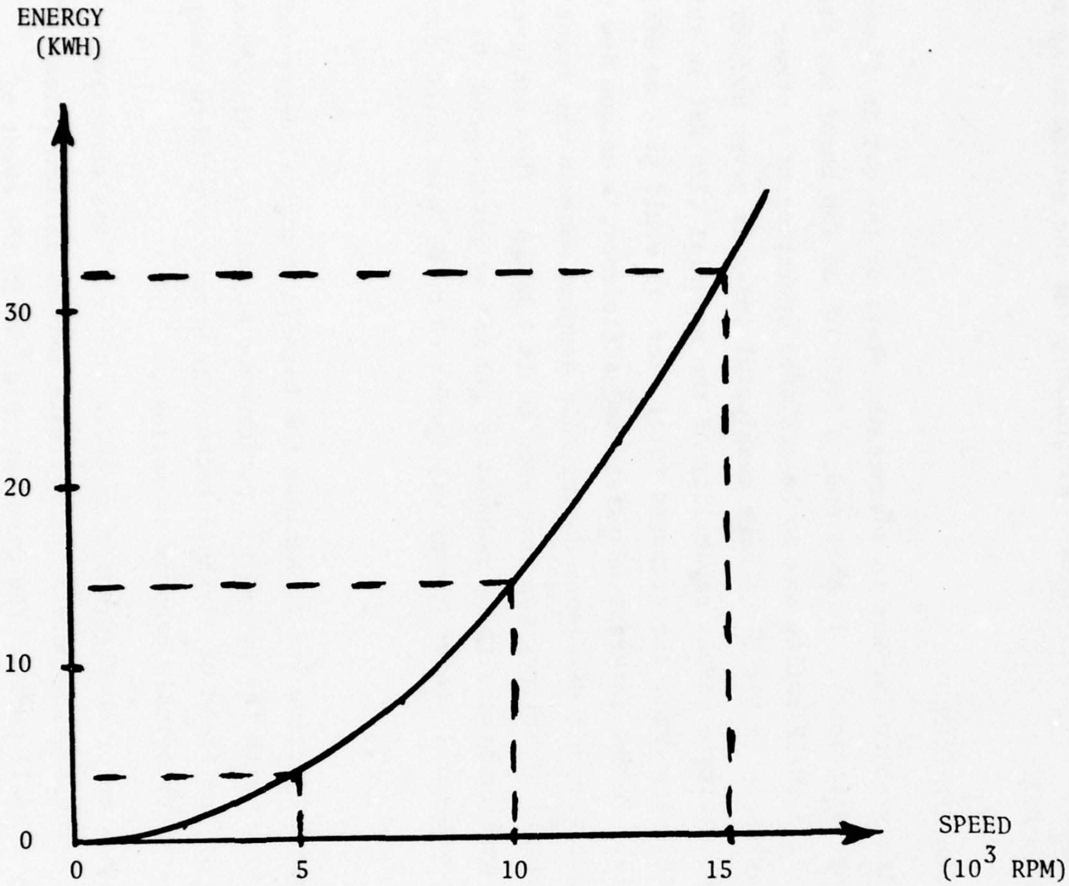

FIG. 1 ENERGY-SPEED RELATIONSHIP

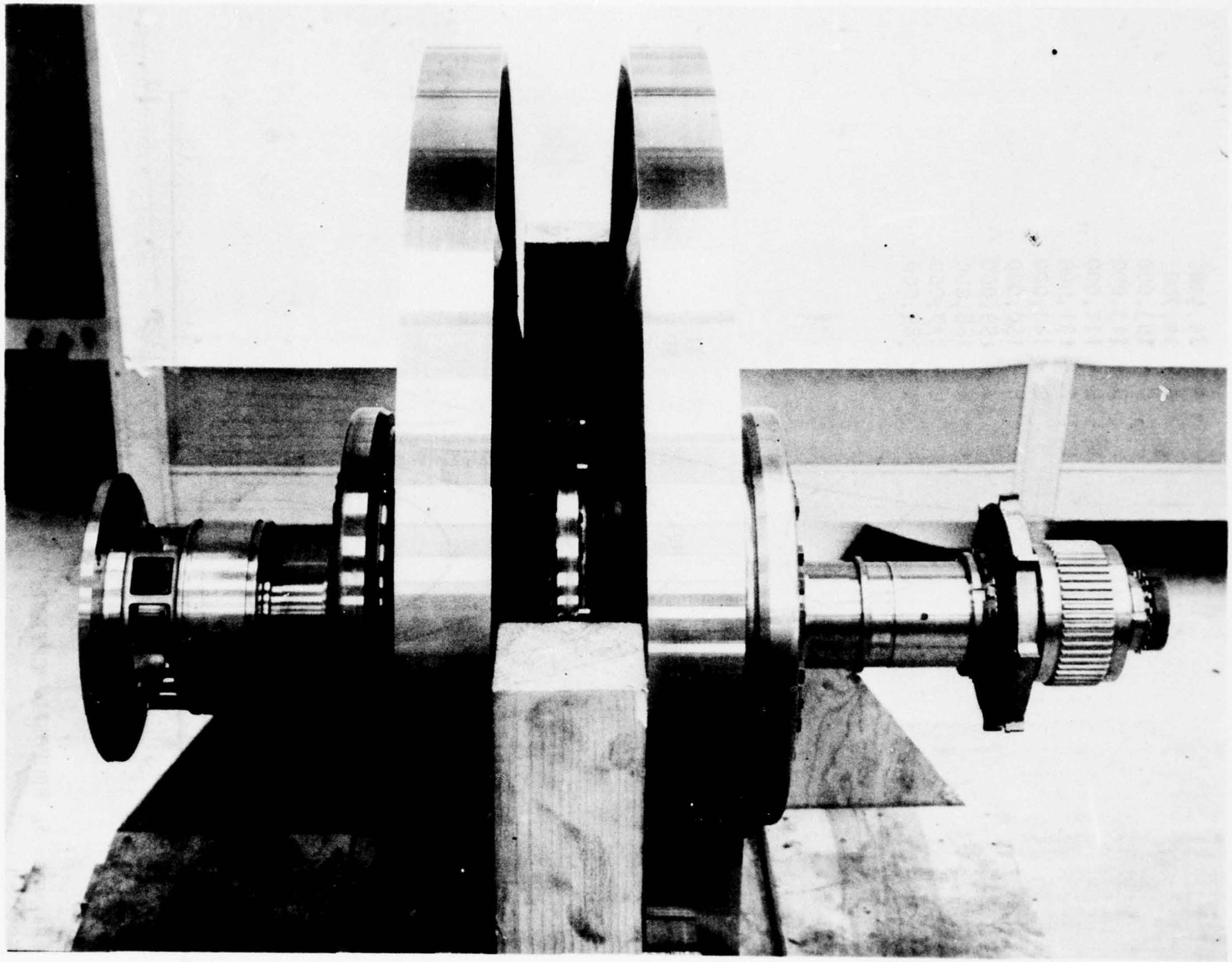

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Figure 2. RS-31 Rotor Assembly

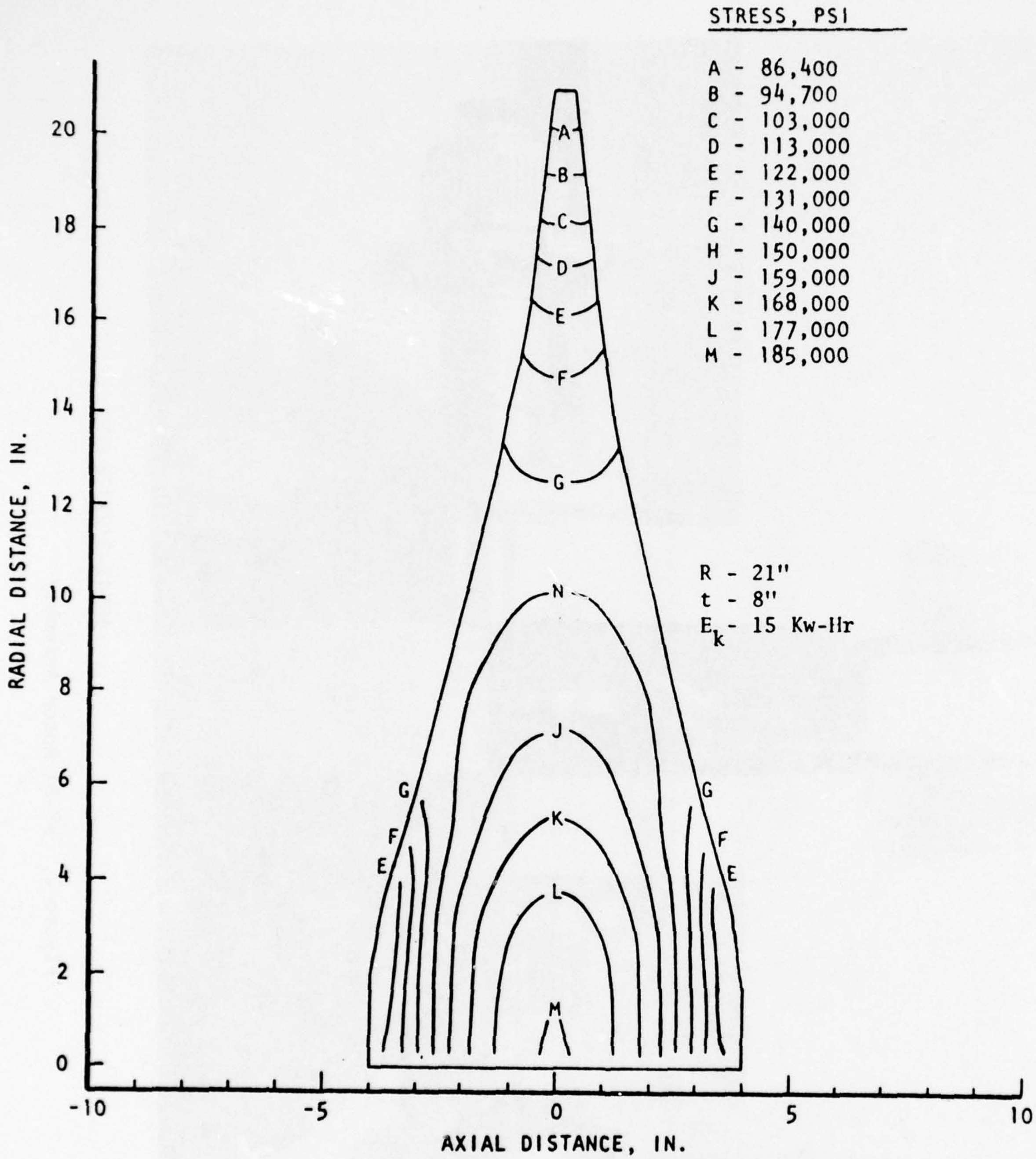


Figure 3. PROPOSED "OPTIMUM" FLYWHEEL DISC

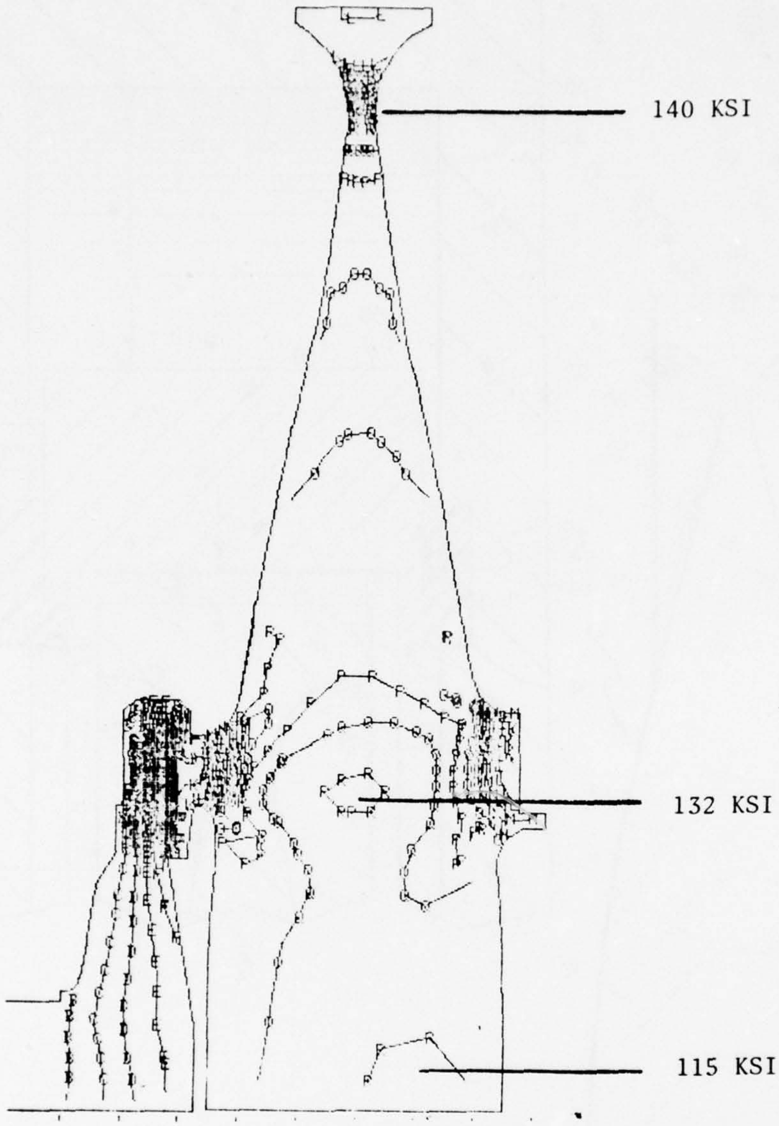


Figure 4. AS-BUILT FLYWHEEL DISC

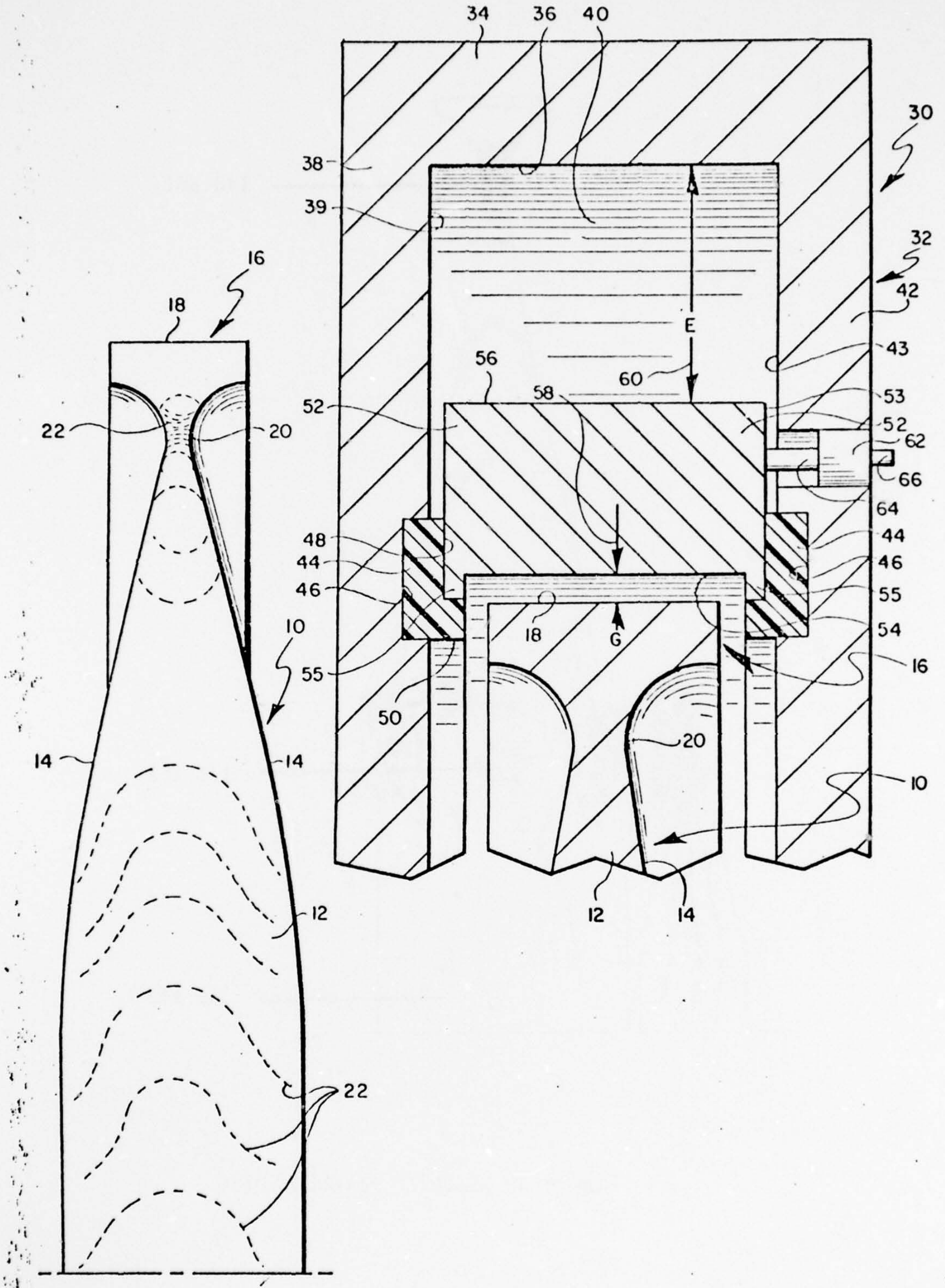


Figure 5. ROTOR SYSTEM CONFIGURATION

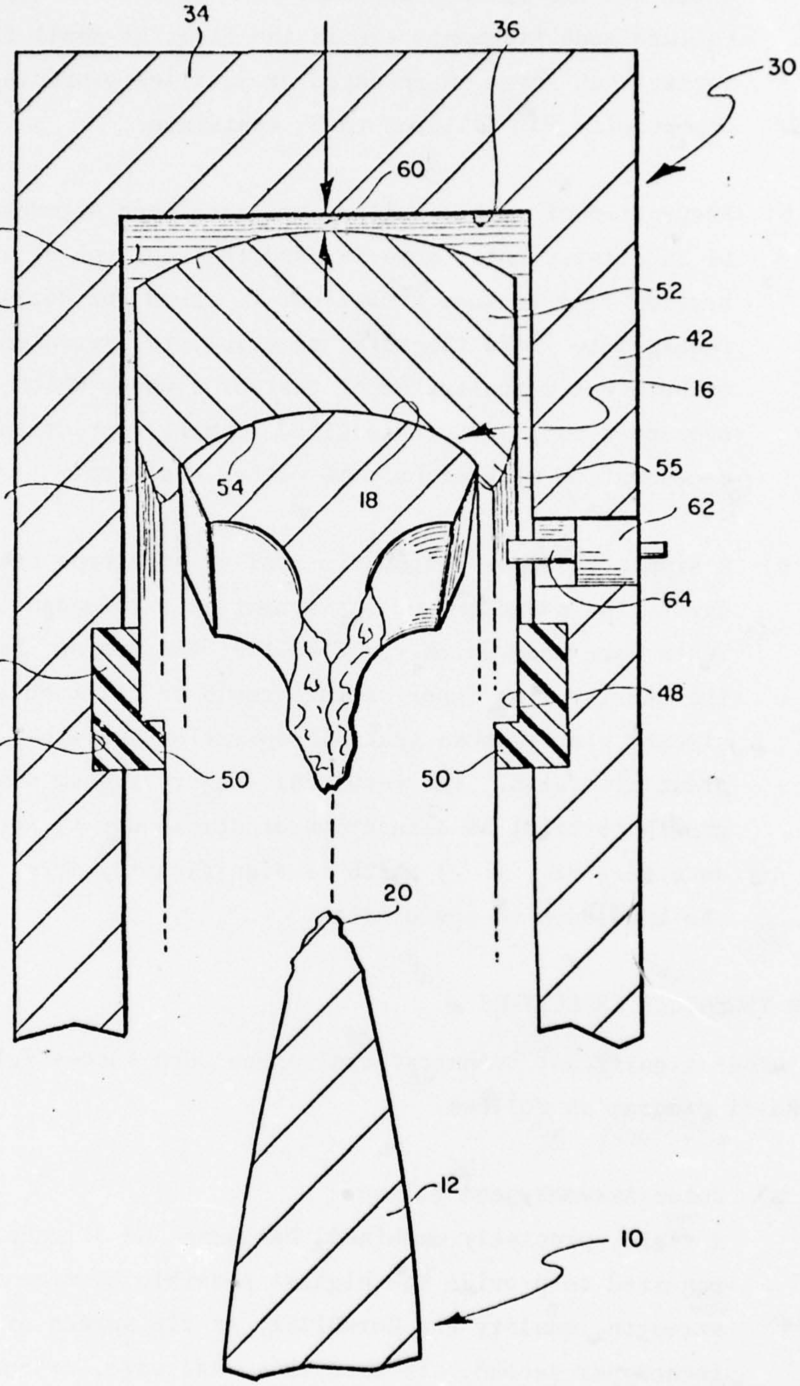


Figure 6 LIMITED FAILURE MECHANISM

This attribute was demonstrated by overspeed of the flywheel until its rim section was shed leaving a clean circular disc. Failure mode fragments are in the forms of small flat plates rather than large sharp-edged projectiles providing a practical opportunity for failures to be contained.

b) Prevention of such a failure has also been augmented by the use of an elastic rotor material and the addition of a precisely fitted barrier ring so that growth due to speed (or heating) in excess of that required to yield the rotor material, is prevented. This attribute was demonstrated by test of a wheel which was allowed to overspeed until growth resulted in ring contact and rubbing deceleration without harm to either component.

c) A simple reliable diagnostic tool of some type other than the first-line speed sensor was sought as a redundant safety device. Tests were made which verified that the rim of the flywheel and the barrier ring inner surface could be utilized as capacitive circuit plates whose spatial separation could be sensed with great precision. The resulting sensory system displays rotor growth (strain) as a function of stress due to speed providing data (See Fig. 7) which is significantly more useful than the knowledge of speed alone.

4. OTHER TECHNICAL OBJECTIVES

Many other significant technical challenges were successfully resolved during the RS-31 program as follows:

a) Rotor Assembly and Balance
A rigid, precisely machined, balanced and aligned rotor is required to provide the highest possible confidence in product strength, quality and durability at tip speeds of up to 31,000 inches per second. To meet this challenge, extensive computer

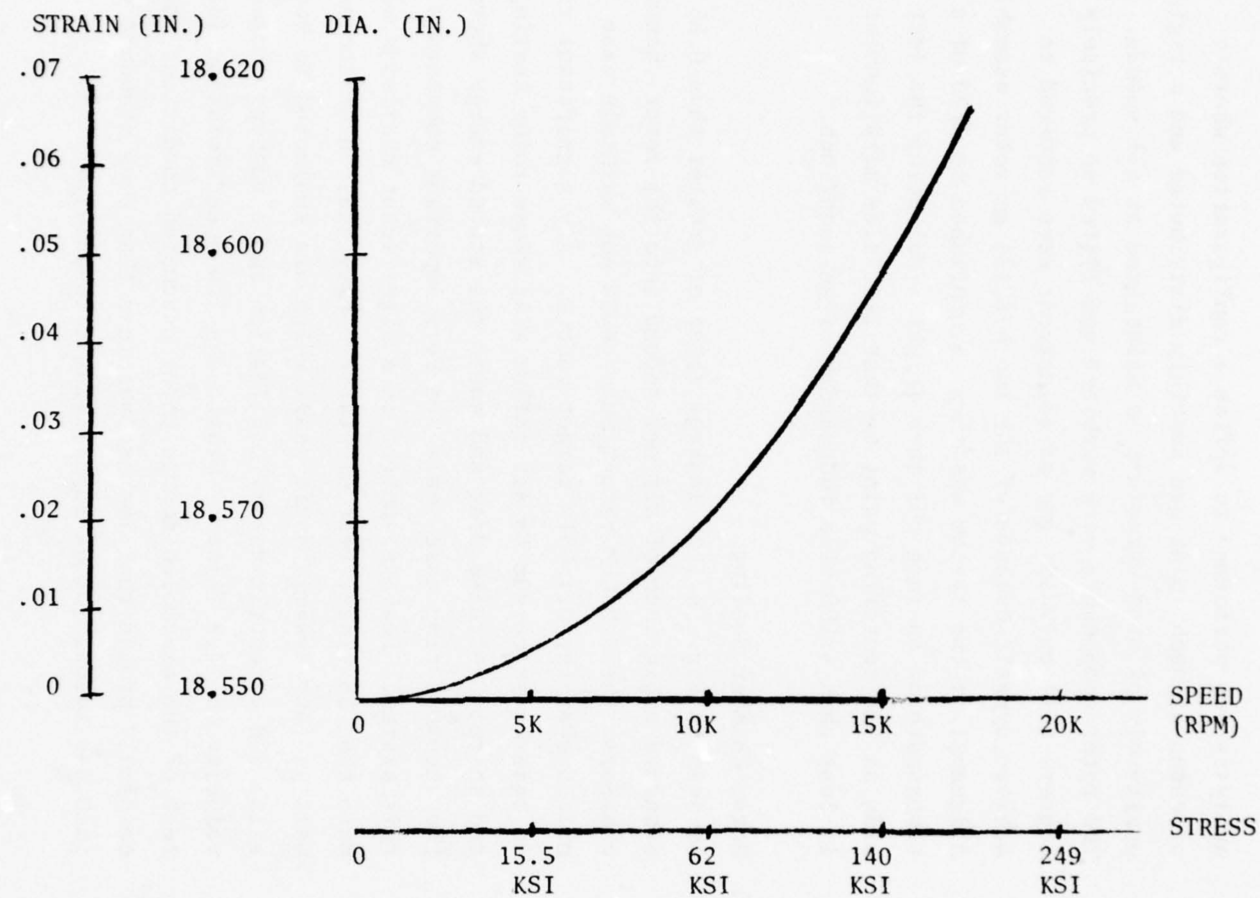


Figure 7. ROTOR GROWTH-STRESS-SPEED DATA

analysis was performed to define a configuration wherein stresses at each joint are smoothly distributed and a rigid positively piloted structure is maintained at all speeds. The rotor components were machined and fitted so precisely on centers that only 18 gms of adjustment were required to achieve dynamic balance of the two 678,121 gm rotor assemblies. Alignment of the system was then accomplished to 3/10 of a thousandth of an inch per inch (right side) using the left side as the centering point so that left side misalignment is less than 5/100 of a thousandth of an inch/inch.

b) Rotor Chamber Sealing

As seen in Fig. 8 , at leakage rates of greater than 0.85 standard cubic inch of air per second into the rotor vacuum chambers, the 4.5 CFM vacuum pump could not maintain case pressure at the 0.1 PSIA target vacuum. Any significant rise in case pressure due to air influx will cause rotor heating and sharply increase drag and waste the stored energy therefore, the vacuum o-ring case seals are very important components of this system. Leakage control is a significant challenge with more than 28 linear feet of static o-ring seal interface as well as four dynamic shaft seals which are subjected to both axial and eccentric rotary oscillation and a rubbing contact velocity of 14.8 ft/sec. Static and low-speed-rotation leak test of the assembled module under evacuated conditions successfully proved that leakage was less than 0.05 standard cubic inch per second (6% of the maximum allowable).

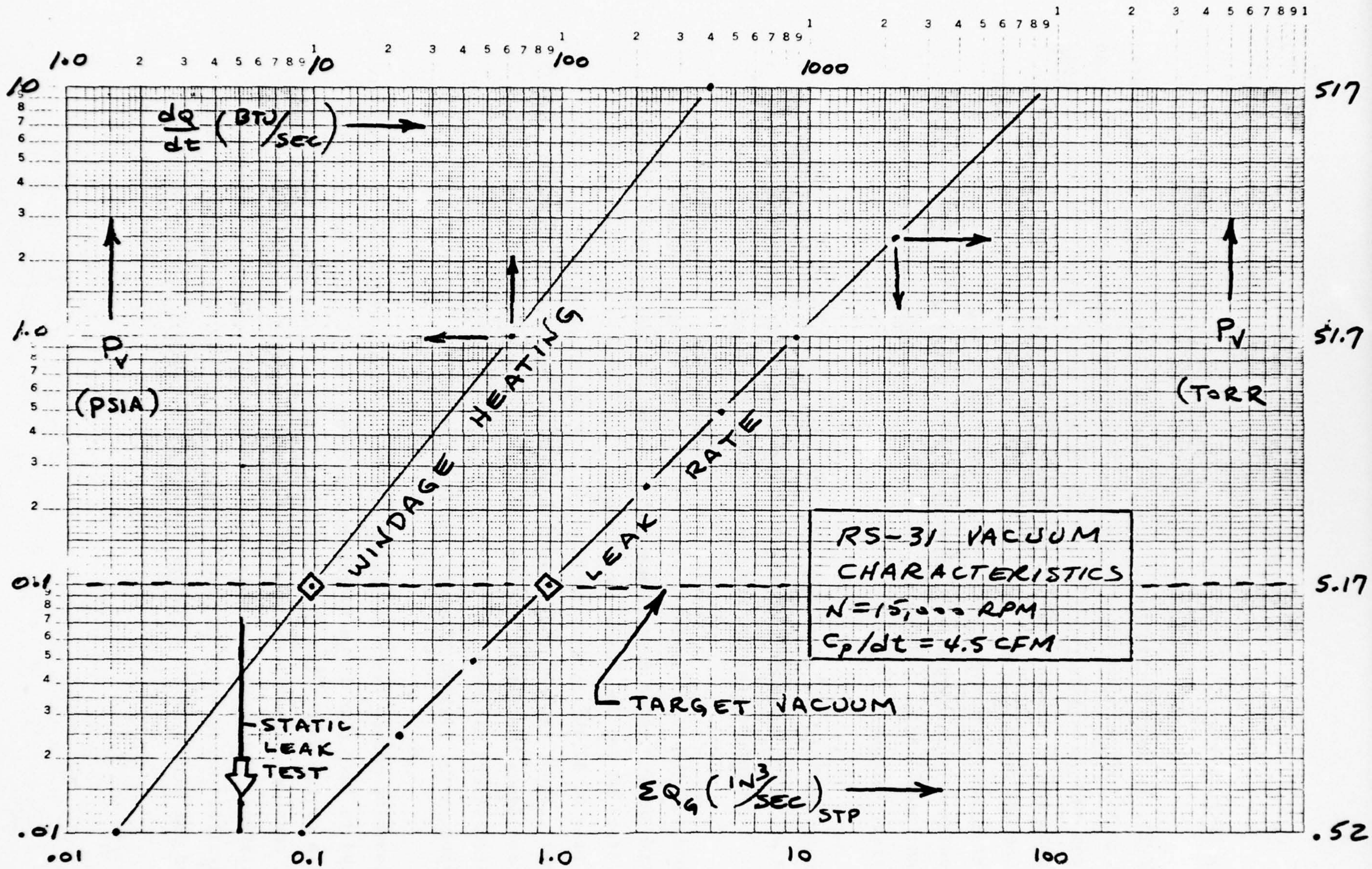


Figure 8. RS-31 VACUUM CHARACTERISTICS

SYSTEM DESCRIPTION

The RS-31, as shown in Fig. 9 , is a Rocketdyne flywheel energy storage system rated at 30 KWH (40 HP-HR) at 14,506 rpm. Accelerating power is supplied by an AVCO gas turbine engine rated at 3000 HP in this application. The decelerating load consists of a pair of Bendix high voltage generators. The load path is from the gas turbine into a four-in-line 1 to 1 ratio gearbox which serves to deliver counter-rotational power to a pair of horizontally aligned parallel flywheel rotors through a pair of sprag-type over-running clutches. At the far end, stored power is absorbed by the directly coupled generators upon electrical command.

Auxiliary controls and instrumentation are included as part of the RS-31 system providing a completely self-sufficient means for evaluation, lubrication, temperature control, speed control and diagnostic data.

The heart of the RS-31 system is its set of flywheel rotors which are machined and assembled from vacuum-melt, forged and heat-treated low nickel alloy steel HP 9-4-30. Each rotor consists of a pair of end stub shafts and a pair of 37.1-inch diameter discs all of which are flanged and bolted together without the concession of disruptive center-disc through-holes. The rotor discs are formed by combining the profile of an infinitely tapered equalized stress disc with an outer rim which improves inertia and stress distribution for limited diametral envelopes.

The assembled rotors (with all shaft rotating elements installed) are each balanced to less than 15 gm-in residual unbalance and mounted horizontally within a casing which also supports the gearbox, gas turbine and generators. Out of consideration for the spatial envelope specified for the RS-31 installation, the casing assembly was designed for an offset placement of the parallel rotors so as to nest the right hand rotor disc pair slightly forward and inside of the diametral envelope of the left hand rotor discs.

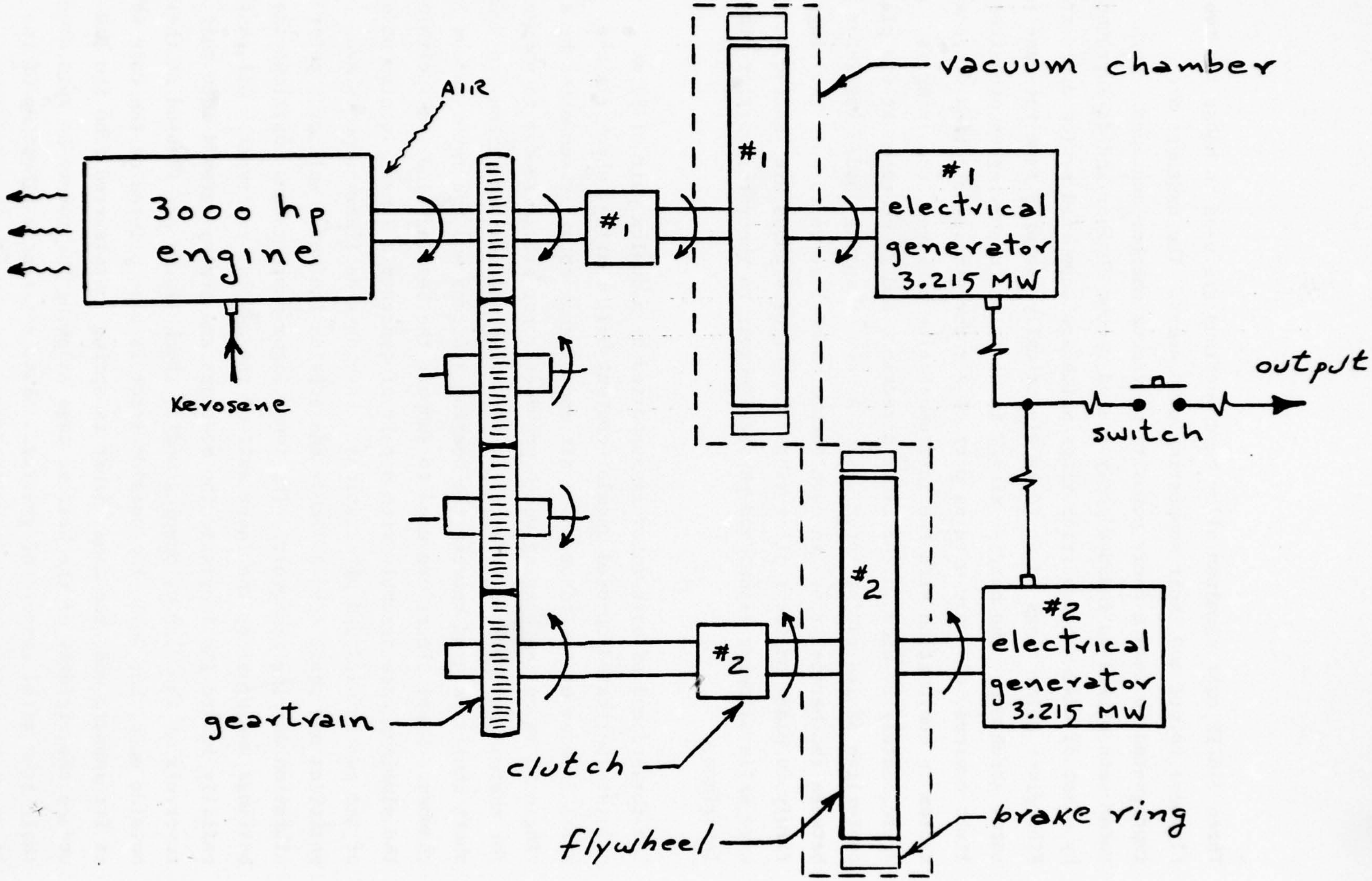


Figure 9. RS-31 Flywheel System

Three 356-T6 cast aluminum alloy case sections are used to house the two flywheel rotors and their supporting components. The castings are impregnated to reduce their porosity as vacuum chamber sections. Each rotor sets within an independently sealed vacuum chamber and is surrounded by a set of 4340 steel barrier rings which are supported by low dialectric fiberglass pilot rings so as to be electrically isolated from the case and rotor structure. The barrier rings, in addition to serving as overstress limit devices, also are used as part of a rotor strain detection diagnostic system in conjunction with the instrumentation package. Each ring is approximately 3" wide in the axial direction, and is centered at the planar centerline of its mating rotor disc. A pair of spring loaded insulator plates between the barrier rings in each vacuum chamber serve to hold the rings firmly in place on their pilot rings. A set of windows are located in the case walls allowing visual and physical access to the rotor/barrier ring interface.

The forward end of each rotor is supported by a duplex pair of 80 mm oil jet lubricated opposed angular contact ball bearings which fix the axial freedom of the rotor. The aft end of each rotor is supported by a single 80 mm roller bearing which provides rotor axial freedom to compensate for thermal and poisson effects. A set of spring loading carbon-lip dynamic shaft seals are used between the bearing stations and the rotor vacuum chamber. Steel liners are used to support the bearings and seals within the aluminum case assembly with a pair of concentric liners located at each of the two forward and aft stations. Each outside liner supports and positions one shaft seal assembly and acts as the outer wall of a rotor vibration damping reservoir. The inner liner supports and positions the bearings and provides the inner wall of the damping reservoir. Oil passing radially inward (to lubricate the bearings and seals) spreads into this reservoir as the viscous damping media. Both liners are flanged at their outside ends, but while the outside liner is also piloted on the case wall at its inboard end, the inner liner is spring cantilevered from the end to reduce the rigidity of the bearing cage allowing the rotors to spin about their true axial centers of gravity. Thus, rotational vibration of the flywheel assembly is not transmitted directly to the case structure.

At the aft end of each rotor a shaft mounted sprag clutch allows the flywheels to be accelerated but when the driver speed is reduced below flywheel speed, the sprags disengage allowing each rotor to spin freely and independent of the driver. A pair of 65 mm ball bearings on either side of each sprag clutch cage serves to separate the inner and outer clutch cylinders. The inner cylinder is splined to the rotor shaft while the outer cylinder pilots the female end of the splined drive shafts.

The aft case of the RS-31 module serves as a central oil collection and distribution center for lubricating oil. One section of the case collects oil from the forward end (externally scavenged), from the aft end by natural drainage and from the gearbox by natural drainage. This oil drains through an internal dearation screen and is piped away for straining and cooling as required. Returning oil is pumped into a sump section of the aft case where a heater allows control of viscosity before the oil is pumped through filters to the bearings, gears, seals and splines.

The AVCO gas turbine bolts to the aft end of the module on the left side of the gearbox and turns clockwise. A splined shaft between the turbine and far left gear of the gearbox delivers drive power to the geartrain. A second shaft which fits into the forward half of this left side gear delivers power directly to the left rotor clutch outer cylinder. Power to the right side rotor is delivered through the four gears of the box so that the right end gear turns counter clockwise. A third shaft delivers power from this gear to the right rotor clutch outer cylinder. The nested rotor centerlines are separated by 28 inches.

A set of eight control packages regulate and monitor module operation. Five of these are standard 19" wide rack panels whose functions are gas turbine control, signal conditioning, limit-alarm processing, power breaker relay functions, and the main operator control and display panel. These five panels are mounted in a standard electrical cabinet. The other three control subassemblies are compact skid mounted units whose functions are oil pumping,

oil scavenging and evacuation and oil cooling. The control system is designed to use 208 volt, three-phase 400 Hz power. A pair of 5.7 HP motors are used to power the oil and vacuum pumps and a pair of 1.0 HP motor driven fans force air through the oil cooler at 2,000 CFM.

The vacuum pump is rated at about 4.5 CFM at 1 TORR. Separate 10 micron filters are provided in each of the three oil delivery mains to forward, aft and gearbox oil jets. These are preceded by magnetic separators and strainers at the sump discharge. An oil level switch in the sump and oil pressure switches in the line as well as dirt alarm switches at the filters allow the operator to be advised of discrepant conditions. Oil supply temperature is automatically regulated at 150°F at all times. The operator's control and display panel provides continuous reading digital display of individual rotor speeds, disc centrifugal and thermal radial growth, case vibration levels, individual chamber vacuum pressure and primary oil parameters as well as alarm lites for several secondary parameters.

Gas turbine and gearbox speed are displayed at the turbine control panel along with turbine exhaust temperature and engine torque.

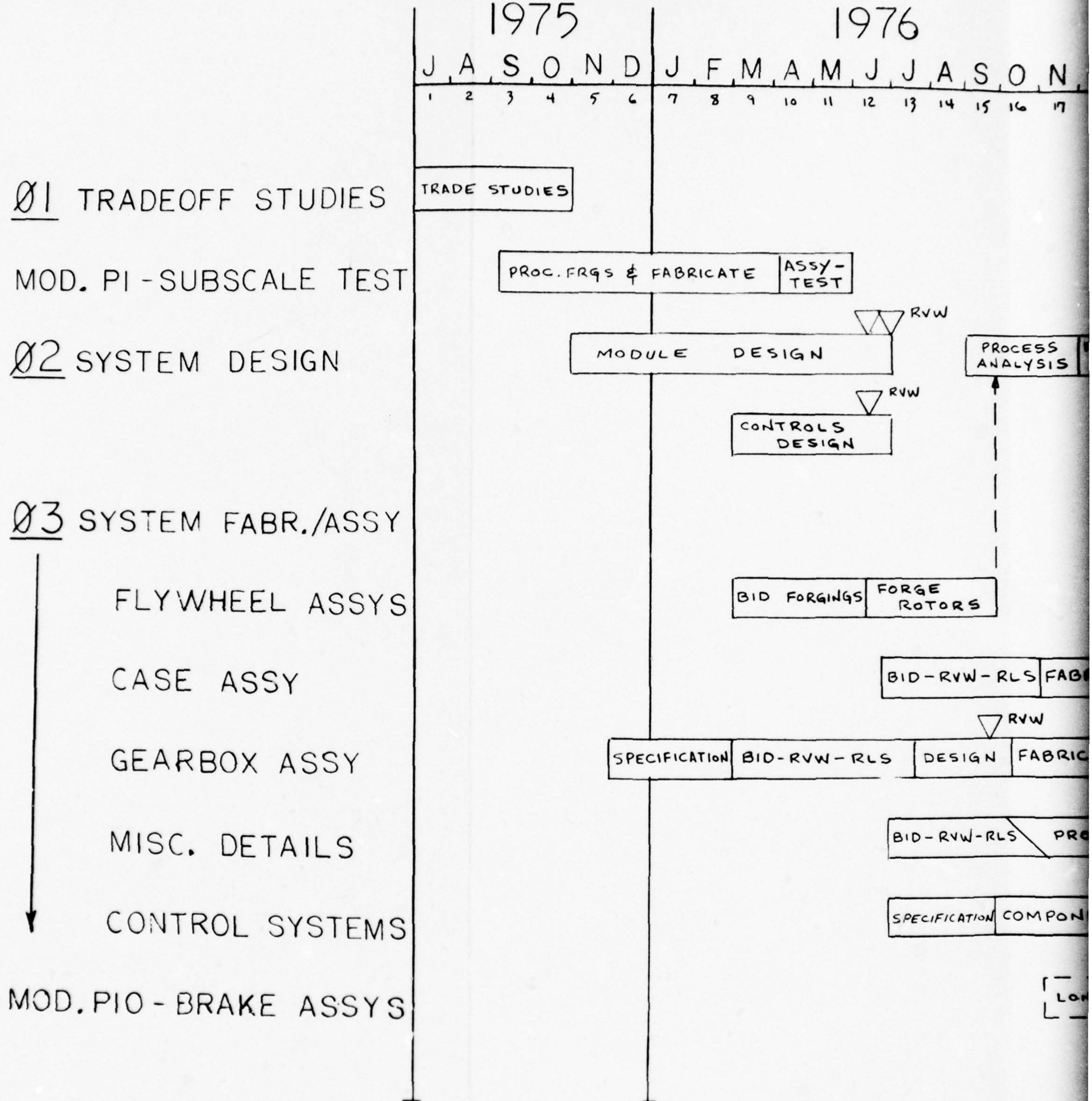
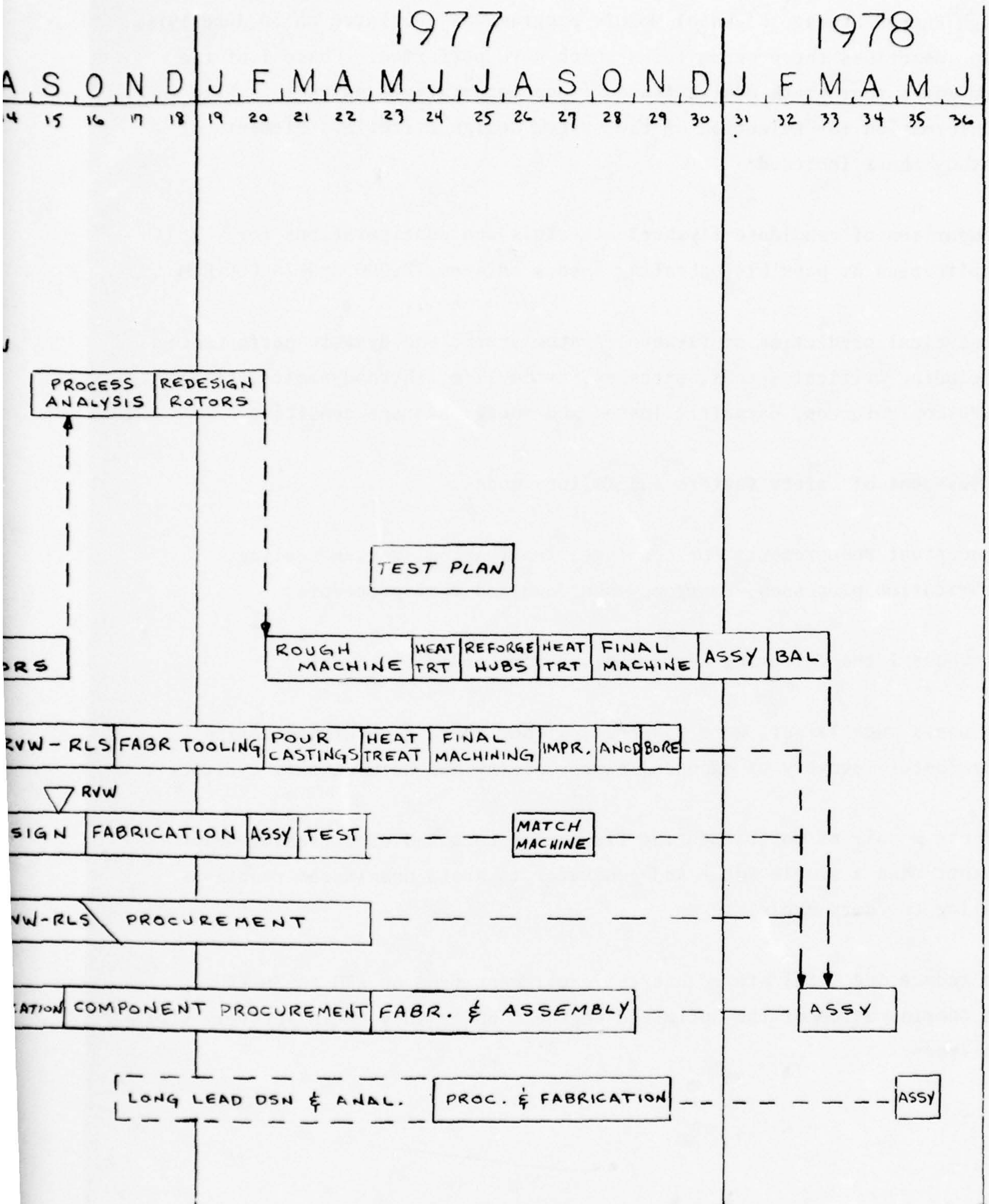


FIG. 10 RS-31 PROGRAM SC



AM SCHEDULE

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PROGRAM DESCRIPTION

The High Energy Storage Flywheel Module program was initiated on 30 June 1975. Fig. 10 describes the program tasks which were performed. Phase I of the program was a four month design tradeoff and comparative analysis task to compile information for selection of the system design criteria. Elements of this study phase included:

1. Comparison of candidate flywheel materials and configurations for application at possible operating speeds between 12,000 and 24,000 RPM.

2. Analytical prediction of flywheel system static and dynamic performance including critical speeds, stresses, cycle life, thermodynamics, balance, gyroscopic forces, parasitic losses and energy storage densities.

3. Assessment of safety factors and failure modes.

4. Conceptual requirements for bearings, lubrication, vacuum sealing, fabrication processes, emergency shutdown and test procedures.

During Phase I the following key decisions were made:

1. To use a much larger, more powerful, higher speed gas turbine engine for faster recovery of expended energy during and between duty cycles.

2. To use a pair of contrarotating flywheel rotors and electrical generators rather than a single rotor and generator to avoid unbalanced reactions during the duty cycle.

3. To reduce the total stored energy requirement from 60 KWH to 30 KWH - in consideration of the optimized engine - power/duty-cycle/recovery-time analyses.

Phase I studies were completed during October 1975 resulting in selection of four 7.5 KWH HP 9-4-30 steel discs (two per rotor shaft) operating at 15,000 RPM. The chief accomplishment during Phase I was a determination that the proposed "optimum" disc configuration could be improved, sharply reducing peak operating stress and thus provide a higher margin of operating safety and cycle life. Also, during Phase I (in September 1975) contract modification P00001 was authorized to add provisions for tests of the selected material and the new rotor disc configuration. Meanwhile, detail system design was initiated under Phase II of the program in November 1975.

As component and assembly design proceeded during the next few months, preliminary drawings and specifications were prepared concurrently for subcontract bids from potential suppliers of the aluminum housing castings, the steel rotor forgings and the gearbox assembly. Severe inflationary impacts of early 1976 were evident in the bids which were received necessitating some adjustment to the program plan and the system design. Accessory drive spur gears were deleted from the gearbox, spare and limits - test rotor forgings were deleted and an alternate aluminum was selected to reduce case material and casting costs. Flywheel rotor forgings were procured during the three month period ending September 1976 and the gearbox was released for detail design in July of 1976,but an intensive four month, 16 source, search for an acceptable (lower cost) case tooling and casting supplier was not culminated until October of 1976. Meanwhile, during early 1976, the subscale flywheel rotor was designed, fabricated and tested demonstrating the fabricability and performance of the rotor design concept. In spite of the unqualified success of this subscale work, a need for revision of the full-scale rotor assembly process plan became evident in October 1976 when occasional weld quality inconsistencies began to appear in production runs on other programs using the required larger size of electron beam welding apparatus. It was determined that the probability of achieving six perfect welds (on two rotors) was unacceptable and a mechanical joint design was substituted during the November 1976 thru January 1977 time-frame. In February of 1977, rotor machining began and the four rotor hubs were delivered to heat treat in May. Subsequently on 3 June 1977, it was determined thru ultra-sonic inspection that three of the four hubs had suffered cracking in the heat

treat process. To augment the probability of success, a more conservative forging design and heat treat process was utilized (rather than the industry standard for this material). Replacement hubs were procured and were then successfully treated. After hub heat treat was successful, a constraint which had been imposed on disc processing was released and disc heat treat success was also achieved using the newly developed criteria. Final machining, assembly and balance of the two rotor assemblies proceeded without incident between September 1977 and February 1978.

Casting tools, in the form of large wooden replicas of the three cases, were completed in February 1977. Sand cores were then completed and the 356-T6 case castings were successfully poured and ready for heat treat by April. Following the final machining phase, each case was impregnated with polyester resin for void-free vacuum-tight walls and the cases were anodized, assembled and line-bored to receive the rotor support structure.

Following critical design review of the Barber-Nichols gearbox in September 1976, fabrication was completed and the supplier subjected the assembled gearbox to no-load test runs to 16,500 RPM during March and April of 1977. So as to assure perfect alignment, the gearbox was left unfinished at the rotor case interface and delivered for match machining to the finished case assembly bore axis. This procedure resulted in excellent final rotor shaft installation alignment as measured in the final assembly.

Long lead procurement of control system components was completed in April 1977 allowing assembly of the four controller panels and three lubrication modules to be initiated.

During November of 1976 authorization was received for the design and fabrication of two rotor air turbine brakes using Rocketdyne's MK-15 hot gas turbine wheels and manifolds as baseline hardware. These brake assemblies are to be used as test equipment in place of the ultimate Bendix electrical generators so that flywheel rotor speed may be reduced more rapidly.

Assembly of the flywheel module was achieved during the month of April 1978 and the brake assembly task was completed early in May concluding the technical program.

The flywheel system currently is in storage at Rocketdyne awaiting implementation of plans for operational testing.

DATA AND CHARACTERISTICS

DESIGN DATA

The following tables of data are provided to summarize characteristics of the RS-31 system.

ROCKETDYNE
LASER PROGRAMS

TABLE I

RS-31 SYSTEM IS COMPRISED OF FIVE MAJOR ELEMENTS

DRIVE ENGINE	-	AVCO LYCOMING (T55-L-7C)
		2930 SHP, 14,750 RPM RATING 7 AXIAL, 1 CENTRIFUGAL COMPRESSOR 2 STAGE FREE POWER TURBINE
GEARBOX	-	BARBER-NICHOLS (RES-1283)
		3000 SHP, 15,000 RPM RATING 4-IN-LINE GEAR TRAIN, 1:1 RATIO INCL 2 SPRAG TYPE OVER-RUNNING, OUTPUT CLUTCH ASSEMBLIES
MODULE	-	ROCKETDYNE (LE76-030-ER)
		32.1 KWH AT 15,000 RPM RATING TWO PARALLEL, HORIZONTAL AXIS CONTRA-ROTATING 2-DISC ROTORS
GENERATORS (2)	-	BENDIX CORP. (28B371-1 & 2)
		3215 KW, 15,000 RPM RATING
CONTROLS	-	ENGINE AND MODULE CONTROLLERS SIGNAL CONDITIONING AND DATA DISPLAYS OIL AND VACUUM SUBASSEMBLIES

TABLE II
GENERAL CHARACTERISTICS

RS-31 MODULE SIZE			4'2" H X 6'6" W X 5'2" L
WEIGHT			7357 LBS
RATED ENERGY STORAGE			30 KWH
PEAK STRESS (DISC)			129 KSI
RATED-ENERGY SPEED			14,506 RPM
STRESS SAFETY FACTOR			1.70 : 1
SIGNIFICANT MATERIALS			
ROTOR	HP 9-4-30	STEEL	
CASES	356-T6	ALUMINUM	
RINGS	4340	STEEL	
GEARS	9310 H	STEEL	

TABLE III
RS-31 CHARACTERISTICS
SIZE AND WEIGHT

AXIAL LENGTH	ENGINE	44.0
	GEARBOX	16.9
	CASE ASSY	45.5
	GENERATORS	31.0
	TOTAL	137.4 IN.
WIDTH	AT THE ROTOR CASE	77.5 IN.
HEIGHT	AT THE ROTOR CASE	49.8 IN.
WEIGHT	ENGINE	750
	GEARBOX	527
	CASE ASSY	2021
	SHAFTS (3) CLUTCHES (2)	75
	BARRIER RINGS (4)	1142
	ROTOR ASSYS (2)	2990
	MISCELLANEOUS DETAILS	602
	GENERATORS (2)	1452
	TOTAL	9559 LB.

TABLE IV
RS-31 CHARACTERISTICS
ENERGY

DESIGN RATED ENERGY - - - - - - - - - - - - - -	30,000 KW-HRS
	40.23 HP-HRS
	80,000,000 FT-LBS
	102,389 B.T.U.
ENERGY AT NOMINAL DSN SPEED - - - - - - - - - (32.08 KWH @ 15,000 RPM)	32.08 KW-HRS
INERTIA OF ROTOR ASSYS (2) - - - - - - - -	320,176 IN^2-LBS
MASS EFFICIENCY OF ENERGY STORAGE AT SPEED	
1. AT RTD SPEED	10.0 WATT-HR/LB
2. AT DSN SPEED	10.7
3. AT MAX SPEED	12.2
4. AT ULT. SPEED	17.5

TABLE V
RS-31 CHARACTERISTICS
SPEED

RATED ENERGY SPEED		14,506 RPM
[30 kWH AT 14,506 RPM]		(1,519 RAD/SEC)
OPERATING RANGE SPEED	MINIMUM	10,500 RPM
	MAXIMUM	16,000 RPM
BARRIER RING BRAKING SPEED		16,500 RPM
MATERIAL YIELD SPEED		17,600 RPM
MATERIAL ULTIMATE SPEED		19,200 RPM
ROTOR EDGE VELOCITY AT MAXIMUM SPEED		1,766 MPH

[SAFETY MARGIN AT MAXIMUM 20%]

TABLE VI
RS-31 CHARACTERISTICS
- PARASITIC LOADS -
(AT 15,000 RPM)

CLUTCH ASSYS (2)	4,268 WATTS	5,7 HP
AFT ROLLER BEARINGS (2)	7,308	9.8
CHAMBER DYNAMIC SEALS (4)	372	0.5
ROTOR ASSY (2) WINDAGE (5 TORR)	5,420	7.3
FWD BALL BEARINGS (4)	8,176	11.0
LUBRICATION PUMPING	4,250	5.7
SCAVENGE OIL PUMPING	608	.8
EVACUATION PUMPING	560	.7
OIL CLR FANS (MAX = 1360 W), MIN →	0	0
OIL HEATER (MAX = 1500 W), MIN →	0	0
GEARBOX (MAX = 23,468 W), MIN →	0	0
	31 KW	41.5 HP

TABLE VII
RS-31 CHARACTERISTICS
STRESS/STRAIN

OPERATING CONDITION	DISC STRESS	RADIAL STRAIN
MINIMUM OPERATING SPEED	68 KSI	0.028 IN.
RATED SPEED	129	0.045
DESIGN SPEED	138	0.048
MAXIMUM OPERATING SPEED	157	0.055
BRAKING SPEED	167	0.059
YIELD SPEED	190	0.067
ULTIMATE SPEED	220 KSI	~ 0.100 IN.
[SAFETY MARGIN AT <u>MAXIMUM</u>	40%	82%]

TABLE VIII
ROTOR SUBASSEMBLIES

- UNIQUE NEW DISC SHAPE PROVIDES
  - 43% LOWER STRESS AT RATED ENERGY
  - STRESS = F (STRAIN) SENSING CAPABILITY
  - PRE-YIELD STRAIN LIMIT BRAKING CAPABILITY
  - BLUNT EDGE FRAGMENTATION CONTAINABILITY
  - 341% REDUCTION IN BURST ENERGY
- 100% ULTRASONIC AND MAGNAFLUX OF VACUUM-MELT, HEAT-TREATED ROTOR COMPONENTS
- ROTOR ASSEMBLIES DYNAMICALLY BALANCED TO ONE-MILLIONTH OF MASS

TABLE IX
ROTOR SUPPORT SUBSYSTEM

- VISCOUSLY DAMPED SPRING LOADED CARRIERS
- DUPLEX BALL BEARINGS FORWARD SUPPORT
- ROLLER BEARINGS AFT SUPPORT
- CARBON FACE DYNAMIC SHAFT SEALS
- 1ST CRITICAL BELOW 5,000 RPM
- 2ND CRITICAL ABOVE 20,000 RPM
- WORST MIS-ALIGNMENT < $\frac{3}{10}$ THOUSANDTH $\frac{\text{INCH}}{\text{INCH}}$
- 25 G RADIAL STATIC BEARING CAPACITY
- 6590 HR COMBINED BEARING FATIGUE LIFE

TABLE X
CASING SUBASSEMBLY

- RUGGED MODULE CASING

REACTS	-	3,000 FT. # CANTILEVERED GEARBOX/ENGINE AFT MOMENT
	-	2,000 FT. # CANTILEVERED GENERATOR FWD MOMENT
	-	1,000 FT. # ENGINE TORQUE
	-	3,000 FT # GENERATOR TORQUE (2)
CONTAINS	-	TWO 44" DIA VACUUM CHAMBERS
	-	OIL SCAVENGE DE-AERATOR SUMP
	-	8.35 GALLON OIL SUPPLY SUMP

TABLE XI
LUBRICATION SUBSYSTEM

- PUMPING ASSEMBLY #032

  PUMP A 2.5 GPM AT 500 PSI
  PUMP B 5.0 GPM AT 500 PSI } AFT LUBRICATION

ROLLER BEARINGS (2)	2.4 GPM
CLUTCHES (2)	2.4
DRIVE SPLINES	1.5
RESERVE CAPACITY	1.2
	7.5 GPM

  PUMP C 2.0 GPM AT 60 PSI } FWD LUBRICATION

BALL BEARINGS (4)	2.0 GPM

- INCLUDES ---------- MAGNETIC SEPARATOR
  0.015 STRAINER
  10 MICRON FILTERS (2)
  CHIP AND DIRT ALARMS
  PRESSURE AND TEMP SENSORS

  [FACILITY BACKUP SYSTEM ALSO PROVIDED]

TABLE XI
LUBRICATION SUBSYSTEM CONT'D

- PUMPING ASSEMBLY #034
  - PUMP D 3.4 GPM FWD SCAVENGE
  - PUMP E 38 GPM AFT SCAVENGE
- TEMPERATURE CONTROL SUBASSEMBLY
  - 1.5 KW THERMOSTATIC CONTROL HEATER
  - 1400 BTU/MIN - °F AIR FAN OIL COOLER
  - AVER. HEAT ADDED TO OIL 32°F
  - HEAT REJECTION CAPABILITY 48°F
  - OIL SUMP TEMPERATURE 150°F
- GEARBOX 0-9 GPM INTERMITTANT PUMPING
  - ADDS < 1000 BTU/CYCLE

TABLE XII

EVACUATION SUBSYSTEM

- 150 LITER PER MINUTE (5.3 CFM) VACUUM PUMP (60% EFFICIENCY AT 1 MICRON)
- TARGET VACUUM 5,000 MICRONS ($\sim$ 0.1 PSIA)
- STATIC LEAK TEST VERIFIED $<$ 0.05 SCI/SEC (INDICATED CAPABILITY $\sim$ 250 MICRONS)
- RS-31 VACUUM SYSTEM BACKED-UP BY 0.33 PSIA HIGH VOLUME FACILITY VACUUM SYSTEM FOR TEST SAFETY

TABLE XIII
GEARBOX SUBASSEMBLY

- 4 CASE-HARDENED 9310H CROWNED TOOTH GEARS
- 30,000 FT/MIN PITCH LINE VELOCITY
- 3,000 SHP TRANSMISSION
- 12 TESTS CONDUCTED (NO LOAD)
  - TO 16,500 RPM FOR 10 MIN.
  - ~ 1 HOUR OPERATING TIME
  - TO 15,000 RPM 10 TESTS

TABLE XIV

FLYWHEEL MATERIALS

MATERIAL PROPERTIES	HP-9-4-20	HP-9-4-30	PHYSICAL PROPERTIES	BOTH
TENSILE ULTIMATE, F_{tu}	190 KSI	220 KSI	DENSITY, ρ	.283 $\#/in^3$
TENSILE YIELD, F_{ty}	170 KSI	190 KSI	DYNAMIC MODULUS OF ELASTICITY, E	28.9×10^6 PSI
FRACTURE TOUGHNESS, K_{1c}	120 KSI $\sqrt{IN}$	90 KSI IN	SHEAR MODULUS OF ELASTICITY, G	11.1×10^6 PSI
PLANE FRACTURE TOUGHNESS, K_c	240 KSI $\sqrt{IN}$	180 KSI IN	POISSONS RATIO, μ	.296
ELONGATION, ε	14-19%	12-16%	COEFFICIENT OF THERMAL EXPANSION (70-200 F), α	6.4×10^{-6} IN/IN-F
HARDNESS, ROCKWELL c	39-44	48-58	ELECTRICAL RE	
			ELECTRICAL RESISTIVITY AT 75°F, ρ	36.6×10^{-6} Ω/Ω -cm
WELDABILITY	BEST	GOOD	THERMAL CONDUCTIVITY AT 75°F, K	170 $\frac{Btu - in}{hr\text{-}ft^2\text{-}°F}$
AVAILABILITY	FAIR	FAIR		

SAFETY CHARACTERISTICS

The RS-31 flywheel rotor has been designed using conservative material properties data supported by Rocketdyne tests. Forged billets of similar thickness were heat treated and adjustments in the process were made until fully hardened conditions were obtained throughout. Tensile and metallographic specimens were prepared and tested to corroborate all handbook data including fatigue and fracture parameters. Precise subscale model flywheel tests were conducted to verify the stress/strain curve, yield point, the failure point and the preferred failure mode.

The RS-31 flywheel disc is designed to fail at its thin neck region just below the rim so that the resulting shrapnel consists of small flat face rim sections. This mode of failure instantly unloads the massive center disc to avoid the classic tri-part mode of disc failure. Subscale tests were carried to destruction of the rotor disc in this manner demonstrating that the central disc that survives in a clean edged, nearly perfectly balanced, tapered disc. Since fragments are small and blunt, a protective barrier ring becomes practical. The barrier ring is also used to provide continuous and accurate safety data by using it as one plate of a capacative circuit. The flywheel's flat rim is the other plate. As the flywheel grows due to thermal and centrifugal conditions the gap between the plates is reduced providing a continuous display of safety margin. This margin of safety is particularly meaningful when it is considered that the preset gap is limited to that permissible before disc yield can set in, therefore, closure of the gap results in a braking effect which prevents degradation of the safety margin to an unacceptably low level. This braking characteristic has also been demonstrated by test.

CONCLUSIONS AND RECOMMENDATIONS

1. The RS-31 flywheel 30 KWH energy storage system has been designed to provide the specified performance at a substantially increased margin of safety over that originally envisioned.
   a) Peak stresses are about 43% lower.
   b) Strain is mechanically limited at 88% of the yield stress.
   c) A preferred mode of failure has been implemented where fragment penetration is reduced by control of fragment shape and energy content.
   d) Rotor strain sensors have been provided for early detection and comprehension of unacceptable operating conditions.

2. The RS-31 flywheel system has been assembled without compromise of a single design parameter resulting in a product of the highest possible quality and safety.

3. Leakage tests of the assembled module and limit speed tests of the gearbox and subscale flywheel all confirm the projections for superior performance of the system.

4. Plans for test of the completed system should be implemented at the earliest practical date.

PHASE I - DESIGN STUDY

Phase I of this program was conducted during the period of 1 July thru 31 October 1975.

MATERIALS

One objective of the design study task was to review and identify all candidate materials suitable for flywheel fabrication, to define all of their characteristics which were significant to fabricability, safety and performance and to select the optimum material.

The initial survey of candidate materials was culminated with a listing of comparative properties of 10 alloys of steel, aluminum and titanium. Then the list was narrowed to Ti-6Al-4V titanium, inconel 718 HP, 9-4-20 and HP 9-4-30 steel. In every respect the HP steels were equal or preferable to the titanium and inconel alloys, therefore, the study was concluded with a thorough analysis and comparison of the two HP steels. HP 9-4-30 was superior in its tensile properties as shown by Table XIV but less well known and somewhat poorly documented, especially in terms of fabricability and thick section properties. Based on the available data, HP 9-4-30 was selected, however, plans were implemented to verify and augment the data with physical test of forged material at an early date. As discussed later, those tests confirmed and justified the selection of HP 9-4-30 as the best flywheel material.

CONFIGURATIONS

ROTOR DISC

At the outset of this design study, disc configurational tradeoffs were expected to be generally limited to diameter-thickness-speed optimizations based on a well documented analytical study of optimum flywheel physical parameters (Ref. 2). This Optimized Stress Computer Model had established that for every material and speed there was a limit diameter (assuming some allowed working stress) and for each diameter/speed parameter there was an optimum thickness. The general shape of the rotor disc was referred to as the Optimized Stress flywheel, essentially a limited variation of an infinitely tapered disc.

The flywheel rotor which had been originally proposed using these criteria was characterized by peak internal stresses to 185,000 psi as presented in Fig. 3. Review of this rotor disc concept gave rise to the following concerns and responses:

1. The 185 KS1 working stress was judged unacceptable in terms of ultimate tensile safety margin (18.9%) and fracture cycle life. A higher safety margin goal of at least 37.5% was initially set (based on 160 KSI working stress).

2. Too much of the disc mass was positioned at a small radius contributing little to the inertia of the disc. Fig. 11 describes an early attempt to redistribute mass and reduce peak stress as finally achieved and displayed in Fig.

3. Location of the peak stress at the internal center of the disc seemed to be asking for catastrophic burst therefore, a fuse or weak-link station near the outer edge of the disc was conceived to promote loss of less than the total mass of the disc from the shaft in the event of overstress.

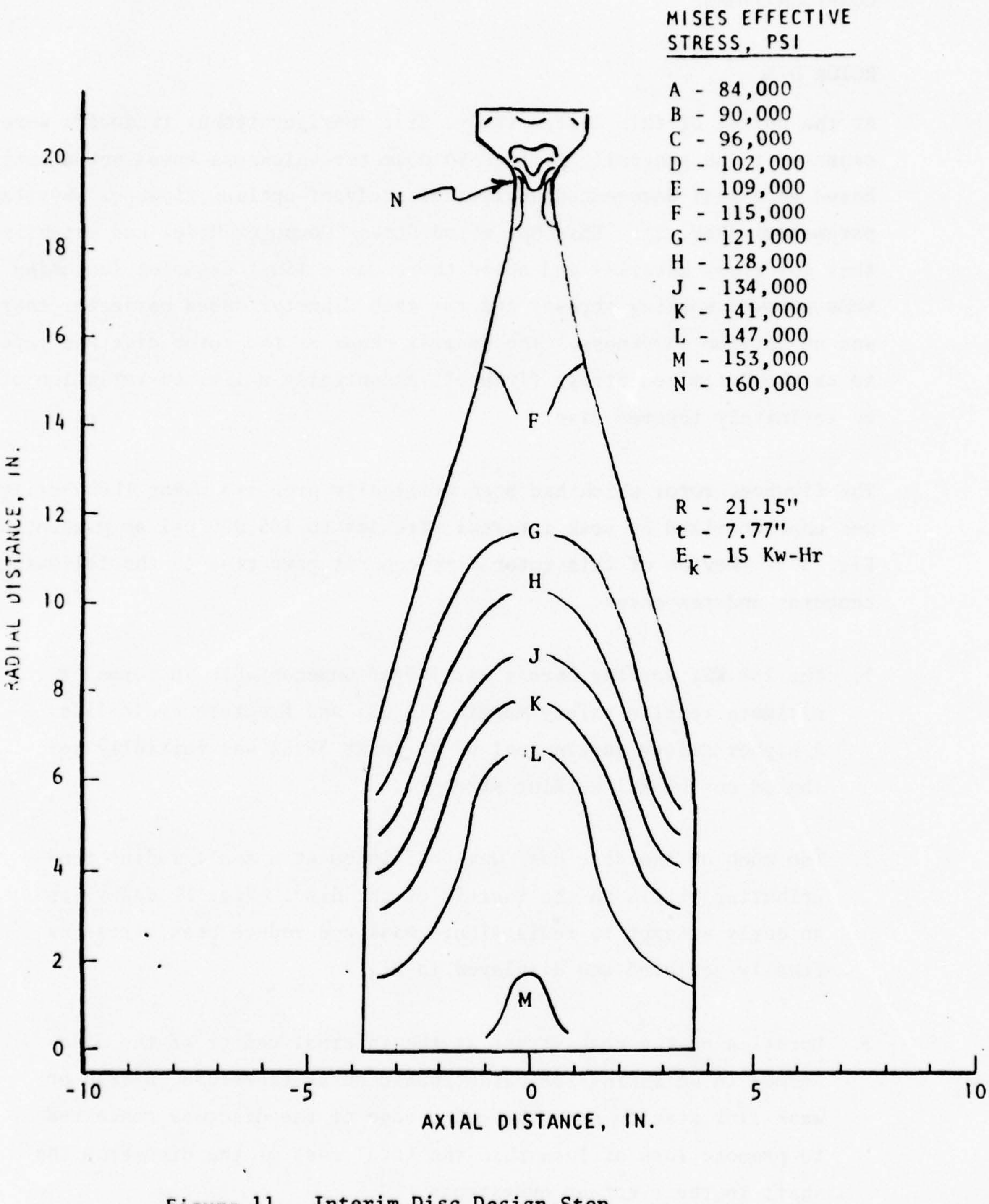

Figure 11. Interim Disc Design Step

4. The sharp edged configuration of the "optimum" disc of Fig. did not seem to be desirable from a containment standpoint therefore the flat edge of the weak-link rim as displayed in Fig. 11 was seen to be a significant advancement in the direction of containability.

As discussed later, under the detail design (Phase II) section of this report, these configuration adjustments were selected and refined to a fine degree by structural analysis to provide a disc of greater stress balance and safety margin.

FLYWHEEL SYSTEM

The specified 60 KWH system was originally envisioned as shown by Fig. 12 with four 42" diameter 16,000 RPM discs on two parallel horizontal contra-rotating shafts driven by a 250 hp Allison model 250-C20 gas turbine (operating at 6000 RPM) thru a 2.67:1 speed increaser gearbox rated at 400 hp (for higher power intermittent duty cycle limit hp capabilities of the turbine engine). Selection of a 3,000 peak horsepower driver (the AVCO T55-L-7C) allowed system energy storage requirements to be reduced to 30 KWH. Additionally, selection of the 15,000 RPM Bendix electrical generators permitted the entire system to be operated at the same speed since the Avco engine was also capable of reaching 15,000 RPM.

A simplified and more responsive configuration was achieved, but case strength and gearbox power requirements were more stringent. The Avco engine weight is 750 lbs (about 5X that of the Allison engine) Support of the heavier 3,000 hp gearbox and engine from the flywheel case increased structural requirements for the case and increased the weight of the overall system. The revised package (Fig. 9) also grew in length because of the need for locating the larger driver engine and the electrical generators at opposite ends of the module rather than nested at one end as had been initially envisioned.

Another improvement in system efficiency, achieved during this study phase, was to provide separate clutches for each flywheel rotor so that the gearbox losses would be unloaded when the driver engine was not powered.

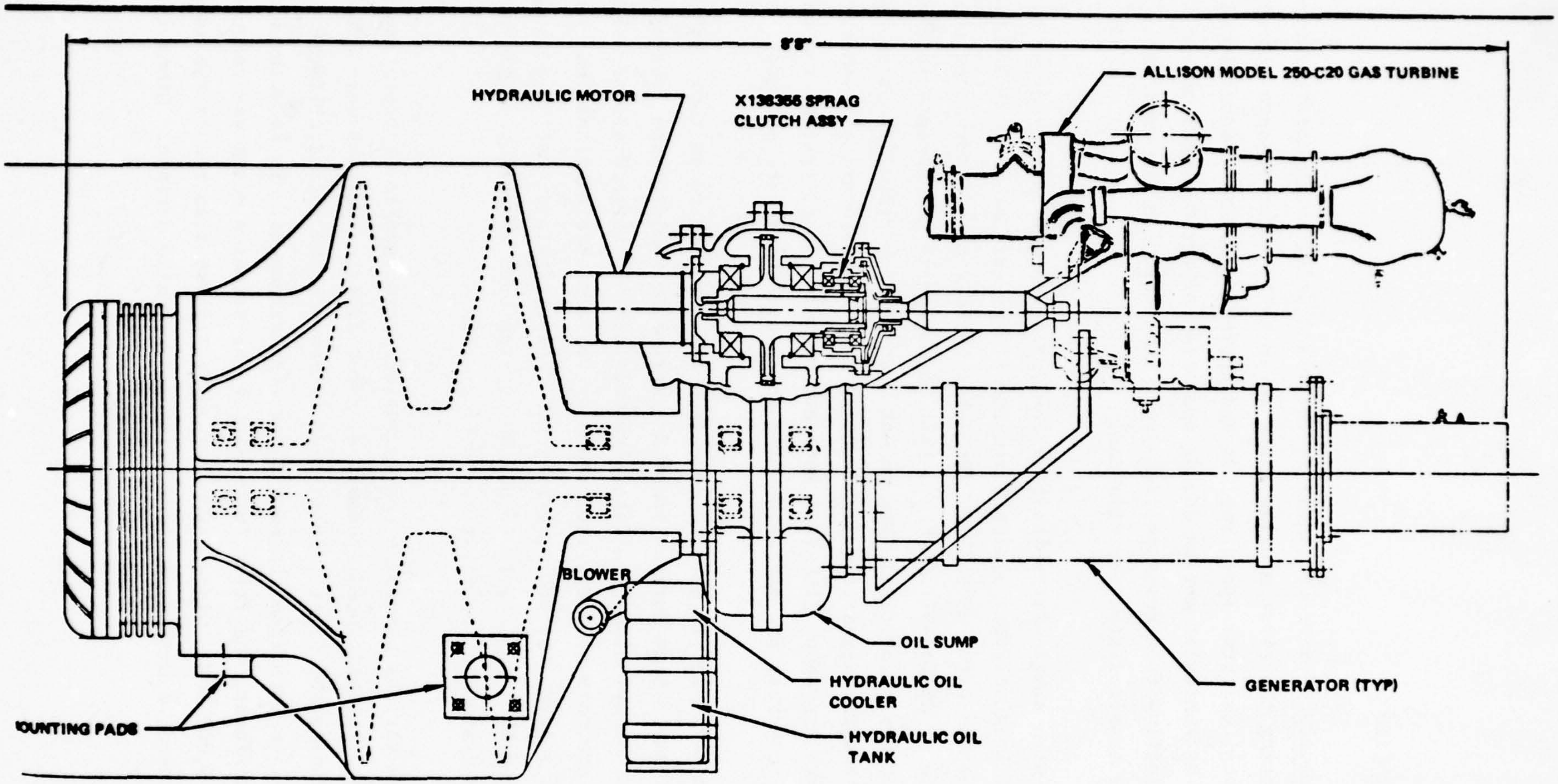

Figure 12. Proposed (Baseline) System

CHARACTERISTICS

System Dynamics of various candidate rotor and drive train configurations were studied during the first program phase to identify rotor balance and alignment requirements, critical speed regimes and operating loads. Processional loads generated by vehicular motion during operation of the flywheel module, and torque resulting from power delivery were considered in the process of developing a stable system installation.

Selection of a single rotor shaft system configuration was ruled out by the results of load delivery analysis because it was evident that extraction of power at the rate of 6.43 MW (8623 hp) could create a torque on the module and vehicle of up to 4311 ft-#. This condition is not acceptable especially since it occurs at precisely the moment when vehicle stability as an optical platform is extremely critical. By using two contra-rotating flywheels whose load torques are about 2155 ft-# each but opposite in direction, the only result is to stress the module housing which can absorb the moments.

If the platform vehicle were to move from one location to another, vehicular accelerations such as surface bumps and cornering would create processional effects on the spinning gyro-like flywheel rotors. Large diameter, slow speed flywheels are more severely influenced by vehicle accelerations than smaller, high speed rotors, therefore, gyroscopic characteristics continued to favor selection of the dual rotor, 15,000 RPM system over competing single rotor 12,500 RPM system in the first tradeoff process.

None of the various candidate configurations were capable of operating below their first critical speed, therefore, rotor dynamic analyses were directed toward reducing the first critical and increasing the second critical speed so as to allow operations between these two resonances. By reducing bearing support stiffness, the first critical for the selected rotor was reduced to less than 5,000 RPM. Since low values of stiffness also reduce the second critical speed, a minimum desired stiffness was also defined. Other influences

on the second critical speed are shaft stiffness (rigidity), end loads, and drive alignment. By careful adjustment of these parameters the second critical was driven upward to about 20,000 RPM. Some of the design methods employed to achieve these results are discussed in later sections of this report under the Phase II activities.

Based upon anticipated state-of-the-art limitations in balance device deflection measurement sensitivity (about 25 X 10^{-6} inches) it was determined that balance of the rotor assembly to about 15 gms-in was a reasonable design criteria (based upon an estimated rotor weight of 600,000 gms).

Thermodynamics of system operations were studied for two reasons. Heat generated during operation not only reduces the allowable working stress of the materials, but directly reduces performance efficiency since stored energy is wasted by the frictional processes which generate the heat. The two most significant potential heating mechanisms are rotor disc windage and bearing viscous friction. Windage losses per disc are proportional to the rotor disc size, speed of rotation, case pressure and fluid viscosity in that order of significance. Case design limitations and rotordynamics considerations tended to dictate that pressure could not be expected to be brought below about 5 TORR and that the optimum rotor size would be 37.10 inches in diameter using two discs per shaft. At 15,000 RPM the dual rotor assembly converts rotor energy to heat at the rate of 10 BTU/sec. Bearings which support the rotors and the clutch assemblies contribute about 19 BTU/sec.

Heat is dissipated by lubricant oil flow thru the bearing areas and by radiation from the flywheel surfaces to the case walls. Three dimensional computer model analysis of the flywheel system thermodynamics served to predict that the rotor disc temperature will ultimately stabilize at 281°F at the outer edge after several hours of rated speed operation. Local outer case wall temperature will reach 222°F at that time as shown by Fig. 13. For this analysis, oil

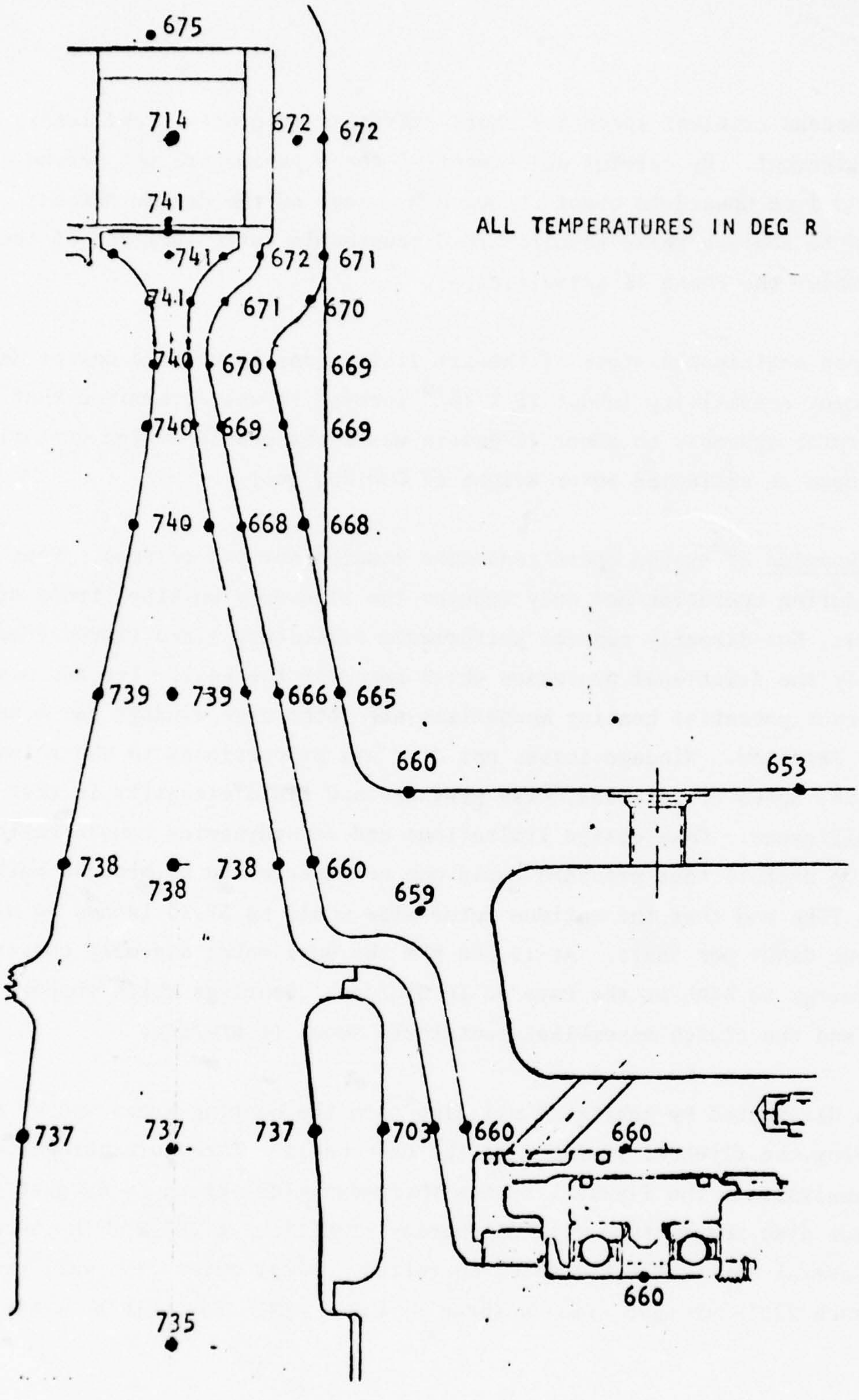

Figure 13. Thermal Model

flow was programmed so as to maintain bearing area temperature at 200°F and case pressure was allowed to reach 7 TORR.

Oil flow temperature control was studied to assure that fluid viscosity would be adequate for long bearing life and an inlet temperature criteria of 150°F was established to achieve oil viscosity characteristics of 10 Cp (IN) and 4.5 Cp (OUT) using MIL-L-23699 lubricant.

Radial thermal growth of the rotor between 70°F and 281°F is 0.025 in. A radial gap of 0.071 inches is provided between the rotors and the surrounding barrier rings, however, rotor thermal growth effects (as shown by Fig. and as determined by subscale tests) are cancelled by thermal growth of the barrier rings. Rotor axial thermal growth of about 0.026 inches at the ultimate steady state temperature is cancelled by poisson shrinkage of the rotor axis (0.028").

SAFETY

The Phase I design study objectives included several critical safety issues as follows:

Potential failure modes of flywheel system operation were predicted so as to identify those aspects of system design which deserved increased attention.

1. Release of the rotor and/or rotor fragment non-containment were determined to be the most hazardous and the most catastrophic modes of system failure.

2. Loss of case structural integrity in any manner which would contribute to loss of rotor integrity or loss of rotor assembly containment was judged to be catastrophic failure mode, but hazardous only in terms of its contribution to the rotor non-containment mode,

3. Loss of bearing system integrity could be a seriously damaging mode of failure but one with low probability of causing rotor failure or ejection. It was estimated that the high energy flywheel will continue to spin about its mass axis within the damaged bearing until the increased journal friction (10 to 20X normal) ultimately brings rotation to a halt.

4. Loss of vacuum in the case was judged to be a potentially damaging mode of failure with low probability of causing a hazardous or catastrophic condition. A 40 to 60 normal windage friction and heating condition will occur at 14.7 PSIA case pressure, but as shown in Fig. 14 the rotor heat increase problem is adequately countered (by reduction in rotor stress) as the windage friction takes its toll on speed.

5. Loss of control, specifically demand for acceleration beyond the safe operating speed of the rotor was judged to be a potentially critical contributory failure mode. Maximum driver engine speed is about 17,500 RPM at essentially zero applied power. At 17,500 RPM flywheel rotor stress safety margin is reduced to 1.17. From the 15,000 RPM design speed, an application of maximum torque from the drive engine for 10 seconds will increase speed to about 16,200 RPM. Since the power turbine shaft cannot exceed 16,200 RPM at the full power setting, any further speed increase to the maximum possible speed of 17,500 RPM would require a reduction in applied power setting.

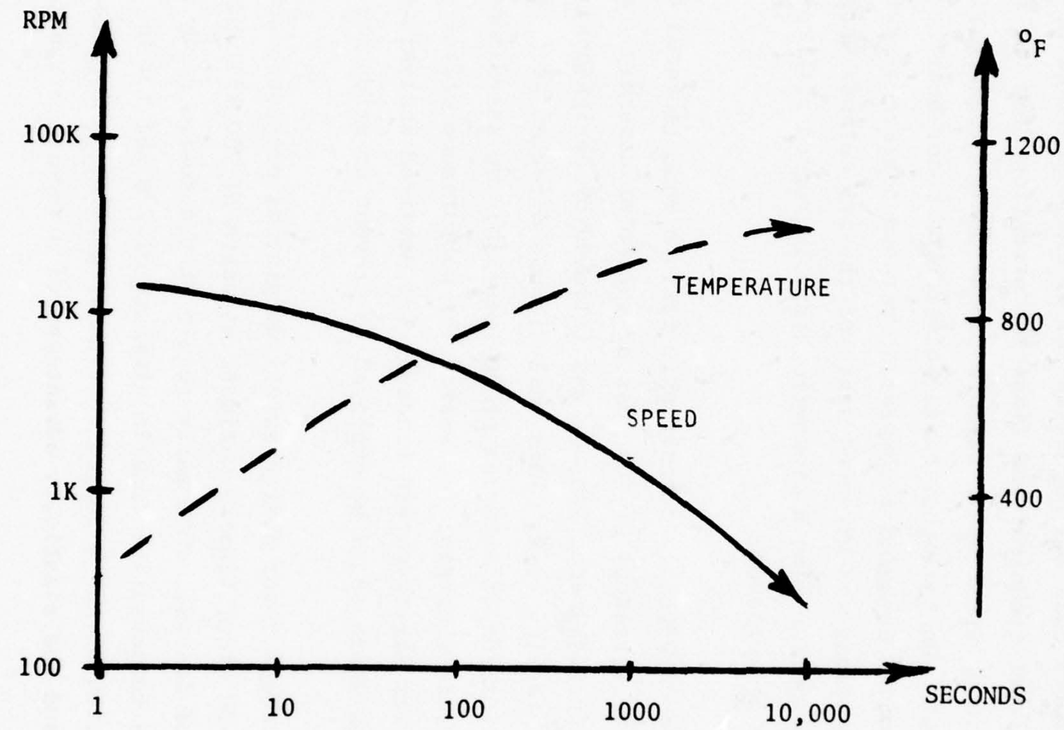

Figure 14. Vacuum Loss Effect on Flywheel Rotor Speed & Temperature vs Time.

The degree to which peripheral elements of the design were technically subordinated depended on the degree to which that component participates in rotor system technology advancement. For example, the case structure safety margin was maximized using the logic that design of a very light-weight rotor case was not essential to the objectives of the program, could divert technical and financial attention from the rotor system, and could cause catastrophic failure if the design logic was, in any respect, erroneous. On the other hand, solution of rotor bearing and seal system requirements demanded a compromise between massive risk-free, low-performance configurations and advanced state-of-the-art methods whose performance could be far superior with a proportionately increased risk to flywheel safety and program resources.

With the only really catastrophic failure mode isolated to (a) loss of rotor integrity (breakup) or (b) loss of the rotor assembly (case breakup), case integrity risk reduction, was followed by design analysis for reduced rotor integrity risk. This analysis was directed at: (a) re-examination and validation of material properties, (b) re-assessment of proposed design safety margin, (c) search for performance efficiency improvement as a path for safety margin tradeup, (d) detailed analysis of protective features which could be employed to prevent or attenuate failure.

The flywheel rotor design safety margin, as proposed, was 1.35; based upon HP 9-4-30 steel tensile ultimate strength of 250 KSI and a working stress level of 185 KSI. One early result of the design study phase was to establish a more conservative tensile ultimate of 220 KSI for HP 9-4-30 in consideration of thick forged section properties, the predicted thermal environment, and the elasticity advantages of a reduced strength heat treat temper.

At 220 KSI the 1.35 margin, as proposed, would allow operation to only 163 KSI. On the one hand the reduced allowable operating stress was an impact on performance efficiency and on the other, studies of rotor disc fatigue cycle life added to concerns that the 1.35 margin was, in itself, immoderate. These considerations intensified the search for design efficiencies which would achieve higher performance at reduced risk and result, ultimately, in a

rotor disc configuration whose rated energy point operating stress of 129 KSI provides a 1.70 safety margin. At the same time the evolving disc design was (a) limited in thickness to improve the quality of construction and non-destructive inspection (b) expanded in edge width to distribute energy per unit projectile area for containability (c) shaped to promote a preferred mode of failure for less energy release.

<u>Damage Potential</u> studies were limited to the overwhelmingly significant case where rotor containment is lost. In the first sub-mode, where case failure allows rotor release as an assembly, conversion of rotational kinetic energy to translational velocity is unpredictable depending upon rotor impact with random obstacles in the area. In the second sub-mode where rotor integrity is lost failure is generally, but not necessarily, characterized by burst of the disc into three parts, each with sufficient projectile energy to penetrate nearly any practical barrier.

With some imagination, however, it is possible to devise barrier conditions which can substantially reduce the damage hazard potential. After some consideration of these factors, a criteria for operations was developed for high energy flywheel systems which included the following tenets:

a) The hazards associated with general use of a flywheel system cannot be totally eliminated anymore than, for example, the destructive potential of an automobile gas tank explosion can be eliminated in some forms of auto crash.

b) Major emphasis should be placed upon reduction of the potential for failure by specification of favorable safety margins, through inspection of the finished product and extensive monitor and control of operating conditions.

c) Developmental test must be performed in a facility designed to attenuate the hazards and damage of failure.

Each of these criteria will be addressed in discussions of the Phase II and Phase III portions of this program.

MODIFICATION P00001 SUBSCALE TEST TASK

As a consequence of recommendations arising out of the tradeoff studies of Phase I, a two-part amendment to the program plan was authorized on 8 September 1975. The objectives of the additional task were to:

a) Augment and verify available technical literature with test data characterizing the properties of the most promising flywheel material.

b) Verify that the proposed new flywheel configuration would perform as predicted.

Starting in September 1975, procurement of forged HP 9-4-30 steel billets was initiated to provide the capability for material sample tests and for fabrication and test of a 14" diameter scale model flywheel assembly.

Material Evaluation

Fabrication Properties of 9-4-30 Material. A forget billet of 1-1/4 x 9-1/2 x 9-1/2 inch 9-4-30 material was used for the tests. The billet was saw cut in the annealed state and machined by engine lathe into a disk 9.005-inch in diameter by 1.135-inch thick. This disk was machined to an approximate 125 rms rinish. This blank was then heat treated (normalized, austenite, refrigerate, temper, and retempered in noncontrolled atmosphere), and was then machined by engine lathe and finish ground on a surface grinder. Steps taken and conclusions reached are as follows:

1. The material was easily machined and cut in the annealed condition.
2. Noncontrolled atmospheric heat treat caused some surface scaling but little distortion, and a growth of +0.015 inch on a 9.00-inch diameter blank. The blank had a Rockwell C-45 hardness after the heat treat.
3. The test specimen disk was machined after heat treat to a 73 rms finish in the engine lathe and 82 rms finish on the surface grinder.
4. The material was drilled and tapped after heat treat.
5. Mill operations, sawing, intermittent cuts should be avoided after heat treatment.
6. A minimum of 0.030- to 0.050-inch stock should be left on the part for finish machine after heat treatment.
7. Low rms finishes after heat treat have to be obtained from hone, lap, or polishing ground surfaces.
8. Heat treat distortion was not a problem unless sections were very thin or irregular.

Hardenability and Mechanical Properties

To demonstrate that the alloy can be fully hardened to the center of a 5-inch section (full-scale flywheel hub thickness) a forging 9-3/4 by 9-1/2 by 12 inches was produced. The forging was annealed, cut to 5-3/8-inch thickness and heat treated. The heat treatment was in accordance with established

Rocketdyne heat treatment procedures. The heat treated section was then cut on the 4-7/8-inch center line and R_C hardness measurements were taken at 1/4-inch increments on the prepared surface. The hardness, R_C 44 , 46, as required, was uniform from surface-to-center as shown in Fig. 15.

Tensile Tests

Smooth and notched tensile specimens and metallographic specimens were prepared from the hardenability test section as shown in Fig. 16. The tensile specimens were oriented in both the longitudinal and transverse directions and used to confirm the minimum design properties, and in addition, the degree of anisotropy that may or may not exist was established. The material toughness was also ascertained from the ratio of notched to unnotched strength.

Based upon an extensive set of tensile tests the minimum material strengths were recorded at 231 ksi ultimate and 203 ksi yield. Exceptionally uniform material properties were recorded, and the notch to unnotch ratio data which averaged 1.53 is unusually good compared to normally acceptable values of 1.10 to 1.20.

Electron Beam Weldability

EB welding parameters were initially determined by bead on plate techniques and then further refined by conventional EB butt welds using appropriate joint configurations and thickness. Visual, radiographic, penetrant, and ultrasonic inspection procedures were used to evaluate weld quality for conformance to Class I requirements.

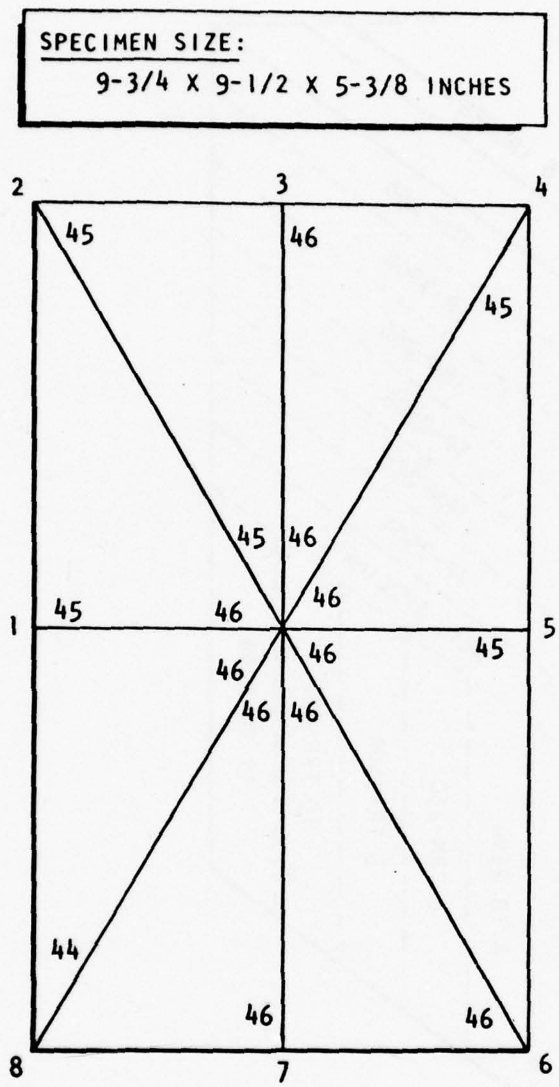

Figure 15 Hardenability Test Specimen Data (R_c Hardness)

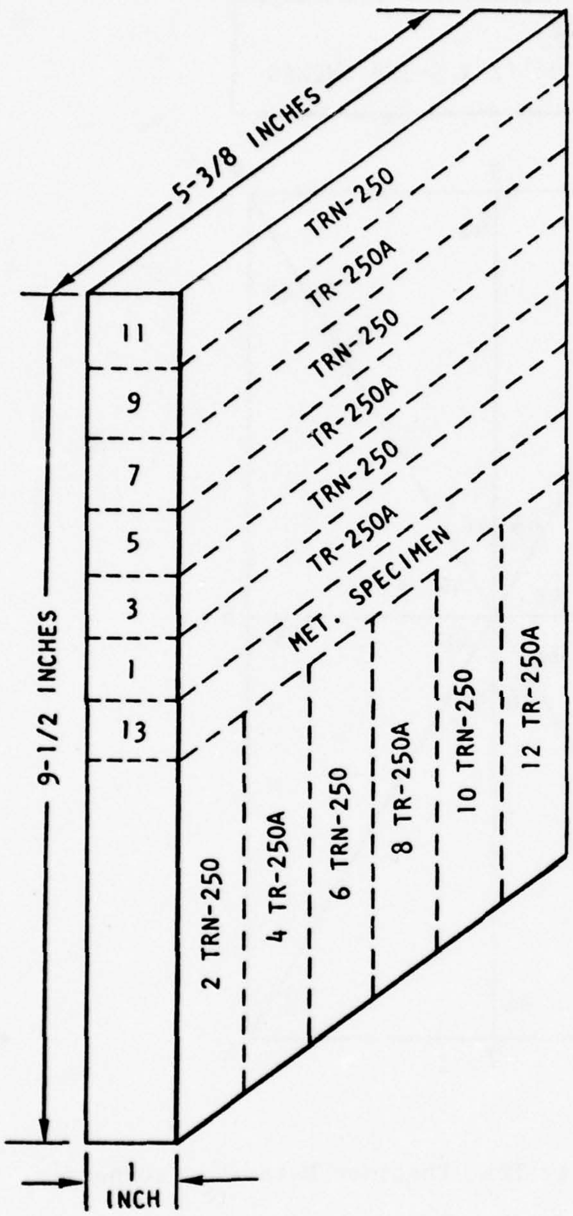

Figure 16. Tensile Test Specimen, HP9-4-30

Material Testing

The high cycle fatigue test program on EB welded HP 9-4-30 alloy utilized standard fatigue specimens machined from lap butt welded plates in which the weld joint thickness was 3/8 inch. The plates to be welded were fully heat treated prior to welding and subsequently were tempered. Weld parameters were those developed by Rocketdyne for HP 9-4-30 alloy in the joint thickness of interest.

The raw data are shown in Table XV , and after reduction are shown plotted in Fig. 17 . Treatment of the data and development of the minimum fatigue values are as discussed below.

The room temperature test data for welded material at R = -1.0 was first normalized to a minimum ultimate strength of 215 ksi and then further reduced an additional 15% to account for the large amount of scatter encountered in the data. The room temperature wrought curve was derived from data generated at the Rockwell B-1 Division and normalized to an ultimate strength of 220 ksi.

The curves for both conditions at 300 F were constructed assuming a degradation in properties that is proportioned to the ratio of the ultimate strengths at 300 and 70 F.

The EB weld fatigue data presented satisfies the weld joint design requirements by providing a fatigue factor of safety of greater than 2 on stress at 10^7 cycles, where the maximum stress curves are approaching a constant value.

TABLE XV HIGH CYCLE FATIGUE TEST DATA SUMMARY

Project:	Axially Test in High Cycle Fatigue 14 EB Welded HP 9-4-30 Thread Hour Glass Specimens
Test Equipment:	MRAI LCF #3 fitted with threaded button adapters BLH Sonntag SF-10-U fitted with an axial test frame
Test Conditions:	Control: Load sine wave Frequency: 5, 6, and 30 Hz Temperature: 70 to 78 F Relative Humidity: 40 to 60¢

Specimen No.	Minimum Diameter, inch	Stress Ratio, R	Stress, ksi			Frequency, Hz	Cycles $\times 10^3$
			maximum	mean	alternating		
P1-1	0.2484	-1.0	130.0	--	130.0	5	10
P2-1	0.2394	-1.0	125.0	--	125.0	5	7
P2-5	0.2521	-1.0	115.0	--	115.0	6	30
P2-3	0.2515	-1.0	110.0	--	110.0	6	12
P1-6	0.2511	-1.0	100.0	--	100.0	30	33
P1-3	0.2510	-1.0	90.0	--	90.0	30	22
P2-6	0.1986	-1.0	80.0	--	80.0	30	276*
P2-7	0.2503	-1.0	70.0	--	70.0	30	320*
P1-4	0.1993	-1.0	70.0	--	70.0	30	7,495
P1-2	0.1991	0.0	106.0	53.0	53.0	30	119
P2-2	0.1984	0.0	95.0	47.5	47.5	30	124
P1-7	0.1981	0.0	90.0	45.0	45.0	30	1,654
P2-4	0.1999	0.7	175.0	148.8	26.2	30	255
P1-5	0.1990	0.7	167.0	142.0	25.0	30	10,290 RO

*Thread failure

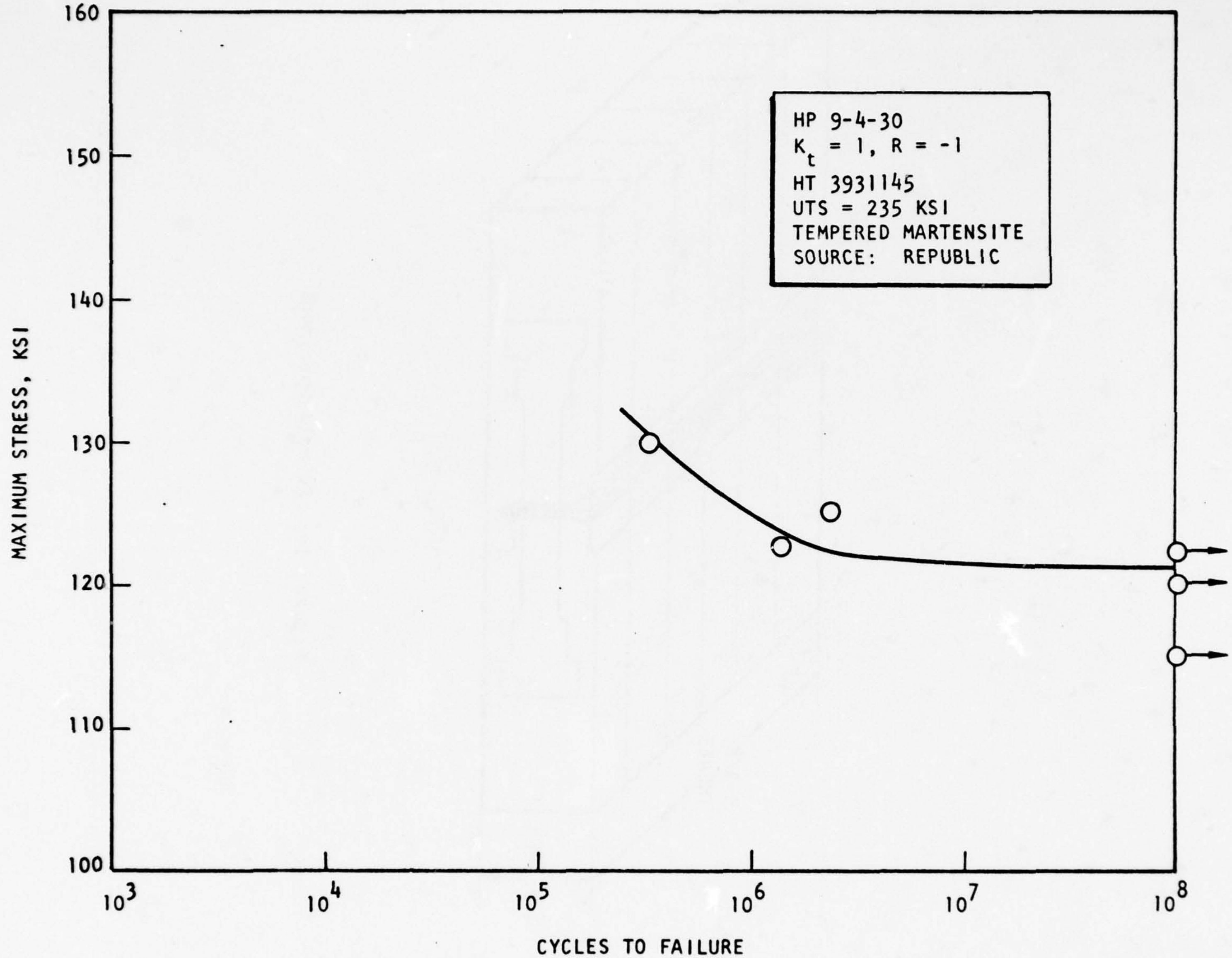

Figure 17. S-N Curve for HP 9-4-30 Tempered Martensite (1-inch Plate)

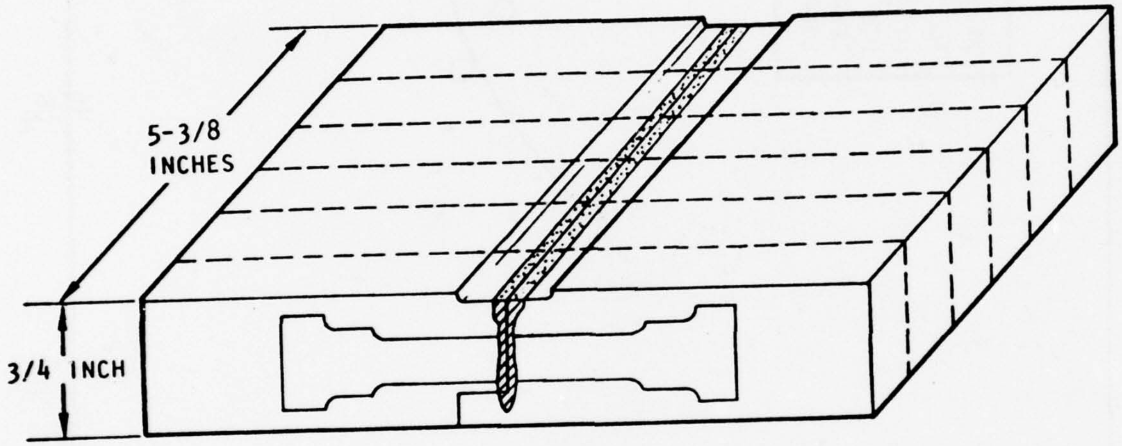

Figure 18. **EB Weld Specimens**

TABLE XVI. MECHANICAL PROPERTIES OF HP 9-4-30 ALLOY-HEAT TREATED SECTION

(Size 9-3/4 x 9-1/2 x 5-3/8 Inches)

Specimen No. and Type	Orientation	Temperature, F	F_{tw}, ksi	F_{ty}, ksi	Elongation, %	Reduction of Area, %	Hardness, Rockwell C	N/UN Ratio
1, Smooth	Longitudinal	RT	234	204	14.0	52.0	47	
3, Notched			360	--	--	--	46.5	1.53
5, Smooth			234	212	14.0	54.5	47	
7, Notched			357	--	--	--	47	1.52
9, Smooth			232	205	14.5	52.8	46	
11, Notched			357	--	--	--	48	1.53
2, Notched	Transverse	RT	358	--	--	--	46	1.53
4, Smooth			233	210	13.0	50.0	48	
6, Notched			361	--	--	--	46	1.56
8, Smooth			231	204	12.0	47.5	47	
10, Notched			358	--	--	--	47	
12, Smooth			232	203	12.0	40.4	45	

TABLE XVII MECHANICAL PROPERTIES OF HP 9-4-30 ALLOY EB WELDED PLATE SPECIMENS

Specimen No.	Condition	F_{tu}, ksi	F_{ty}, ksi	Elongation, %	Reduction of Area, %	Hardness, Rockwell C	Failure, location
1	As welded	229	184	11.5	47.2	47	Heat affected zone
2		222	179	10.5	48	47.5	
3	1000 F for 1 hour	223	196	11.5	47.6	44	
4		220	195	10.0	49	44	
5		233	200	12.0	47	47	
6	1000 F for 2 hours	224	195	12.0	44	43	
7		231	196	13.0	43.8	45	
8		230	204	12.5	46.8	46	

*Double tempered 1 + 1 hours

Subscale Flywheel

System Description. The 14-inch diameter subscale rotor was designed to demonstrate satisfactory performance of the full-scale (37.1-inch dia.) rotor profile at precisely equal stress levels. A 3 dimensional scale reduction of the full-scale wheel was subjected to the same structural analysis computer routine as the full-scale wheel with rotor speed increased until identical stress levels were displayed (at 39,750 RPM). A set of HP 9-4-30 forgings were then procured in accordance with full-scale material processing criteria and the parts were machined to their ultimate profile. Final assembly of the subscale wheel was accomplished by electron beam welding of the disc to each end shaft. Subsequently, studies predicted that EB weld assembly would not be an optimum process for the larger and much heavier full-scale flywheel however, results at the 14-inch size were excellent.

Tests of the assembled subscale flywheel were conducted at Rocketdyne using an air turbine driver in a vertical cylindrical vacuum tank. The flywheel was suspended with its axis vertical using a 5/16 inch diameter shaft from the air turbine. An aluminum housing was fabricated to surround the flywheel and a 4340 steel brake ring was fitted within the housing for two purposes:

a) To demonstrate the capability of limiting speed and thermal growth of the flywheel at a point lower than the yield point of the rotor material.
b) To demonstrate that the rotor growth may be precisely measured during operation so as to provide material properties verification and functional safety data to the operator.

As discussed in other sections of this report, the RS-31 system design includes a brake ring which provides both of these features. The growth sensing characteristic is achieved by using the rotor ring and the closely fitted brake ring as parallel plates of an electrically capacitive network which defines

rotor growth as a function of gap variation to provide an indication of stress levels in the disc.

The assembled flywheel rotor was dynamically balanced and then a series of five spin tests were conducted to achieve the task objectives. The first two tests were low speed runs to about 10,000 RPM during which adjustments were made to the installation to add a lower guide bearing, align the axis of rotation and calibrate the instrumentation. Subsequently three tests were performed.

Objectives. The primary test objectives were:

a) To demonstrate satisfactory operation to rated speed (39,750 RPM) and to demonstrate the constraining action of the brake ring at 120% of rated speed (47,700 RPM).

b) To obtain permanent deformation of the rotor by operation in the plastic range at speeds of up to 51,675 RPM without burst thus demonstrating a 30% safety margin.

c) To demonstrate precisely controlled operation to burst at the predicted failure speed of 53,663 RPM (35% above the design rated speed) and further to specifically demonstrate the design failure mode of rim separation without disc failure.

All of these objectives were demonstrated as described below.

Test Results.

Test No. 3 was significant in the following respects:

1. The 39,750 rpm design rating for the rotor was validated.
2. A substantial margin of safety was demonstrated by carrying the rotor speed 20.15% above the design rated level without rotor failure or degradation.
3. The rotor gap detect system (which were designed to fulfill requirements for an alternative fail-safe monitor of speed

limitation) provided precise data on rotor growth versus speed and triggered the FASTAX camera at contact as preprogrammed.

4. The rotor brake ring (which is designed to prevent over-speed) provided limit speed control when contact was made at 47,760 rpm.

Figure 19 describes the rotor speed profile versus time for test No. 3 as well as the concurrent temperature data at the lower bearing support plate near the inner race. The lower bearing was added to facilitate acceleration of the wheel without wobble at low speed until polar-moment-of-inertia serves to stabilize the rotor. After ring contact was achieved, speed decay was augmented by the application of reverse torque from the air turbine.

Comparison of the measured growth data of Fig. 20 with the material properties and stress analysis prediction indicates almost exact agreement from 15,000 to 47,760 rpm. Figure 20 includes gap detect data versus speed for both the acceleration and deceleration cycle. The gap circuit trigger was set to actuate a camera. This set point corresponds to more than 0.0350-inch growth (50,000 to 51,000 rpm) or contact (which saturates the detector circuit). The contact was detected properly triggering 18 seconds of movies covering rotor operation from peak speed of 47,780 rpm down to about 47,600 rpm.

The rotor ring impact was clearly evidenced in the movie by shocks to the assembly support structure as well as by gap variations visually detectable in the local area of vision at one edge of the rotor.

The wheel was examined posttest showing a rub mark over a local area of the outer rim face about 5-inches long circumferentially. The hardness reading in this area was found to have increased from the machined value of 50 to a local hardness of 55, slightly increasing local strength and brittleness, but of no significant consequence to further test. Penetrant inspection was passed and the rotor was returned for test No. 4. The ring also showed local rub marks

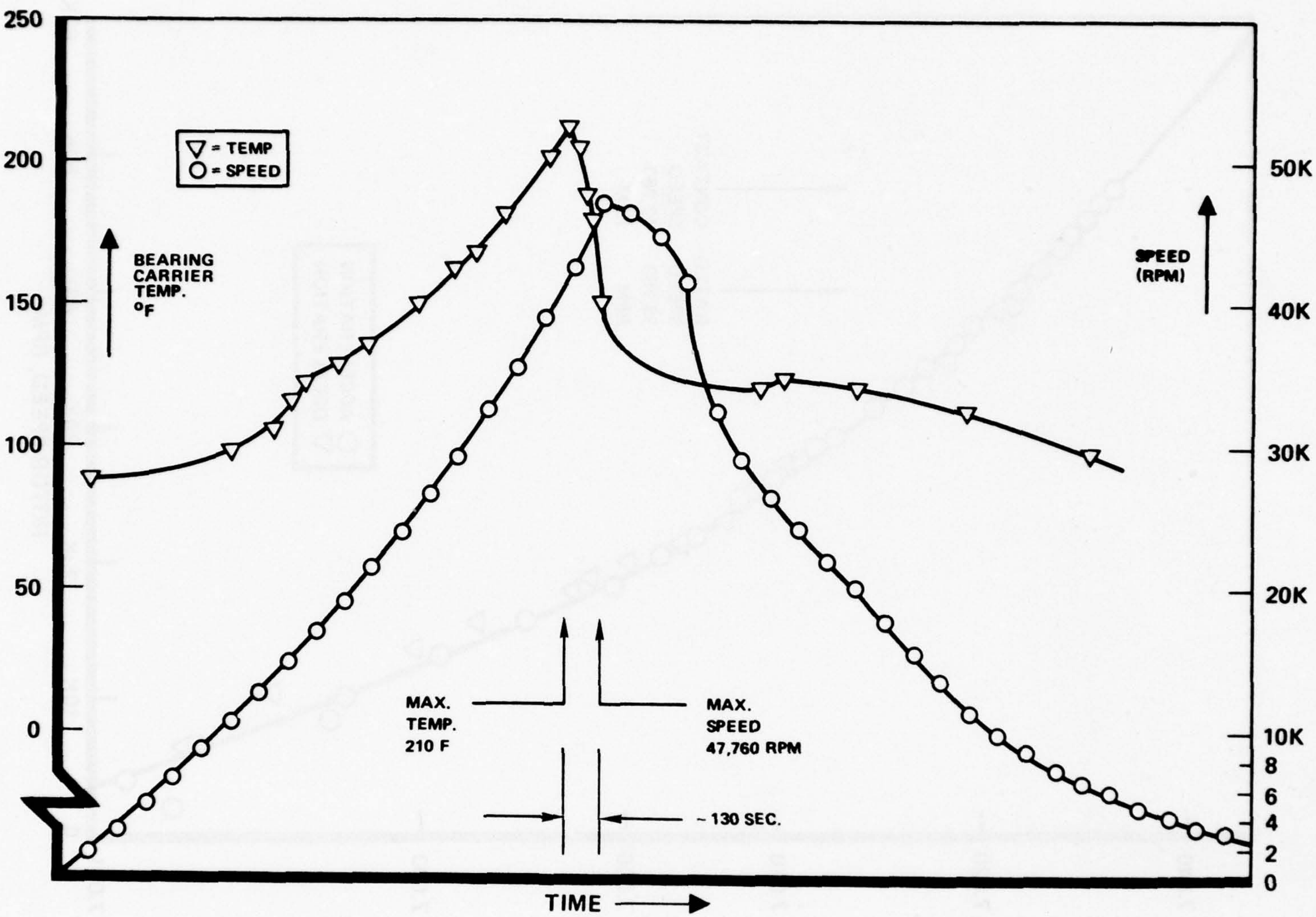

Figure 19 Test No. 3 rpm vs Time Profile

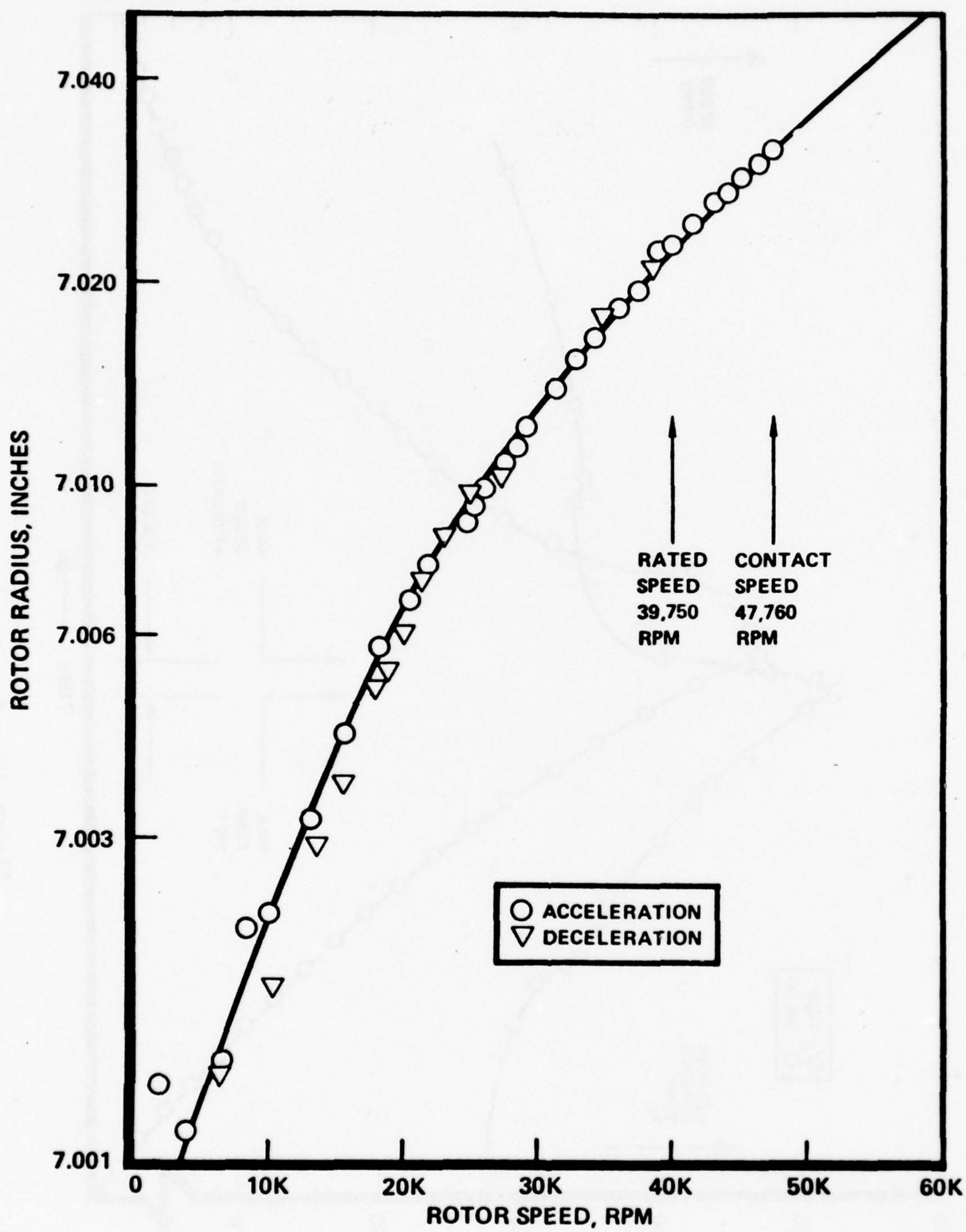

Figure 20. No. 3 Test Rotor Growth vs Speed

in two adjacent areas of about 5-inches each around its 14-inch ID. Both parts remained within their fabricated tolerance of ±0.001-inch radially with no evidence of rotor permanent deformation. The ring was then resized to 14.120-inch ID preparatory to test No. 4.

Tests No. 4 and 5 were conducted on 26 May 1976 using a new lower bearing selected for the higher operating speed and longer test duration which was planned.

Figure 21 describes the speed profile of tests No. 4 and 5 versus time. During test No. 4 speed was momentarily brought to 51,060 rpm at 38 minutes and then held between 50,000 and 50,500 rpm for 7 minutes. During this period, rotor-to-ring gap was recorded and compared to elastic and plastic growth predictions shown in Fig. 22 , and it was concluded a small permanent deformation at about 0.0002-inch had been encountered. The possibility of uneven thermal growth influence was considered in estimating whether or not yield had occurred. As a check, speed was then reduced to 45,460 rpm at 48 minutes. Figure 23 shows the traverse up and back to 45,460 rpm. Rotor radius enlargement was recorded at all reduced speeds relative to the initial upramp. From 46,000 rpm, rotor speed was increased to 51,500 rpm at 57 minutes, where further yield was evident. After the test, rotor measurements were taken verifying that the pretest gap was reduced indicating growth. During acceleration, windage heating causes the rotor and the barrier ring to heat. Thermal studies indicate that the long term effect is that both parts come to the same temperature, but during transients, the ring grows faster and shrinks slower than the rotor because of its insulated condition, and the disk heat dissipated to the hub and shaft. The gap detect sensor data as presented here also are corrected for thermal growth. The data of Fig. 23 identify that first yield occurred at 50,000 rpm. A re-acceleration from 40,000 rpm seems to pass 50,500 rpm and reach 51,000 rpm before further yield is evident. In subsequent test No. 5 the data indicate that speed was brought to about 52,000 rpm before the growth rate accelerates.

In Fig. 24 the speed increase to rotor failure is shown between 100 minutes and 121 minutes 16 seconds. Rotor growth was displayed on a digital meter and

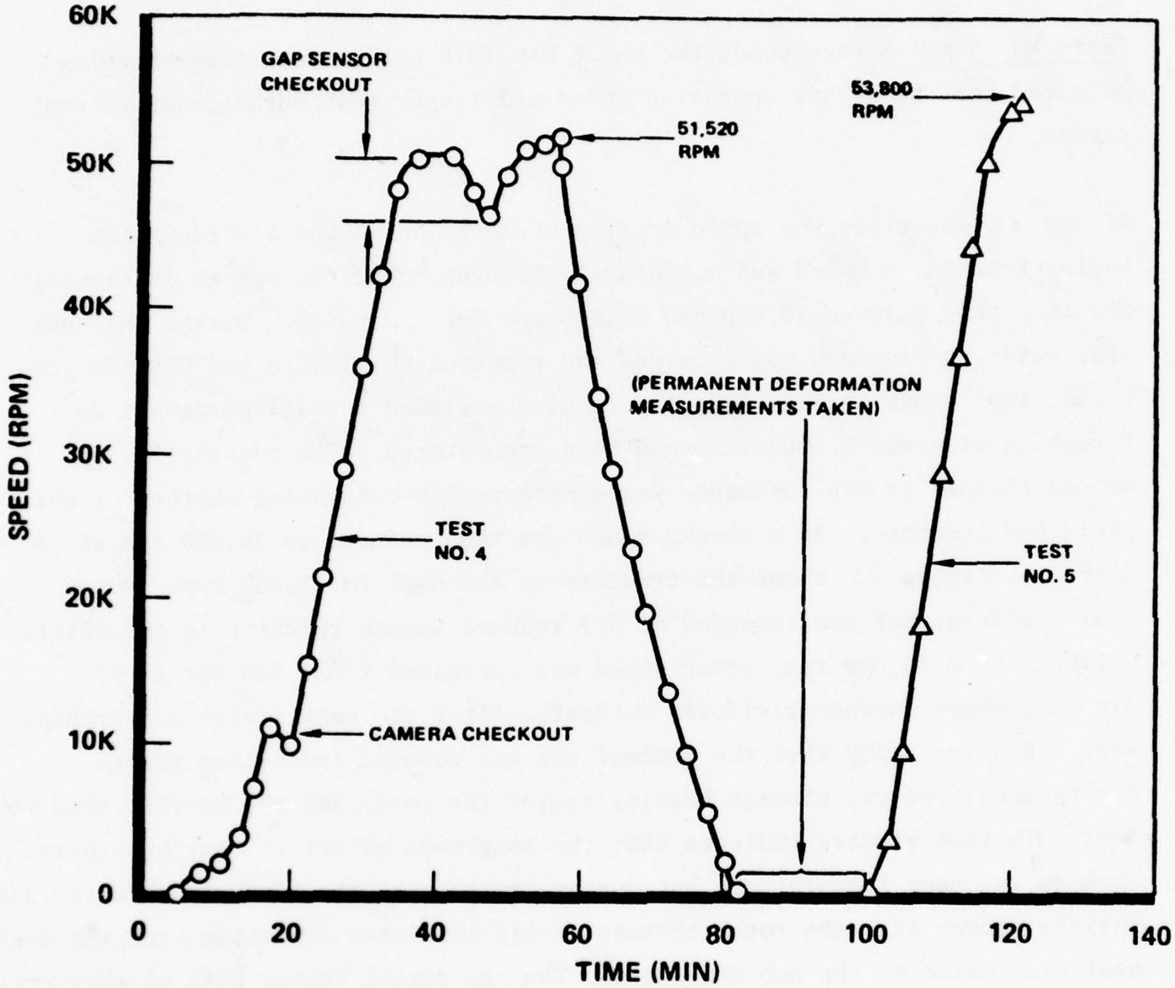

Figure 21. Tests 4 and 5, Speed Profile vs Time

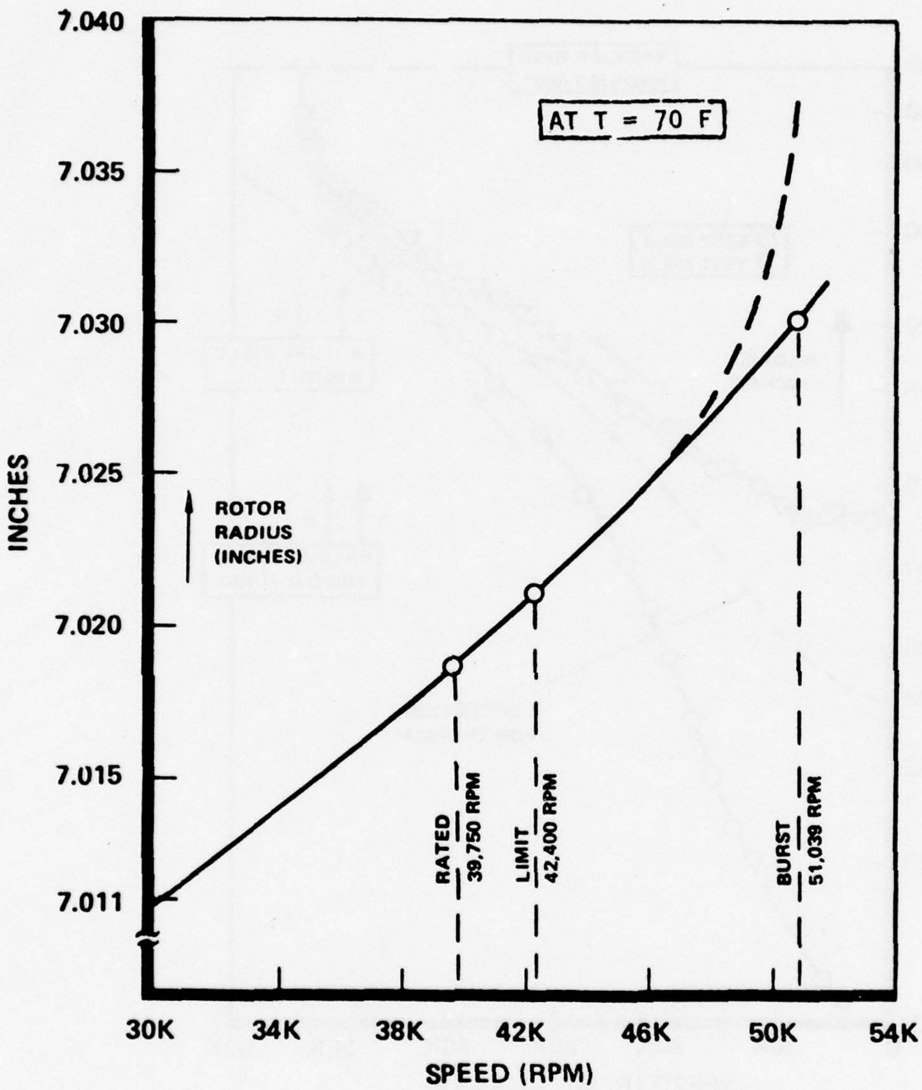

Figure 22. Subscale Rotor Theoretical Stress Prediction of Radius vs Speed

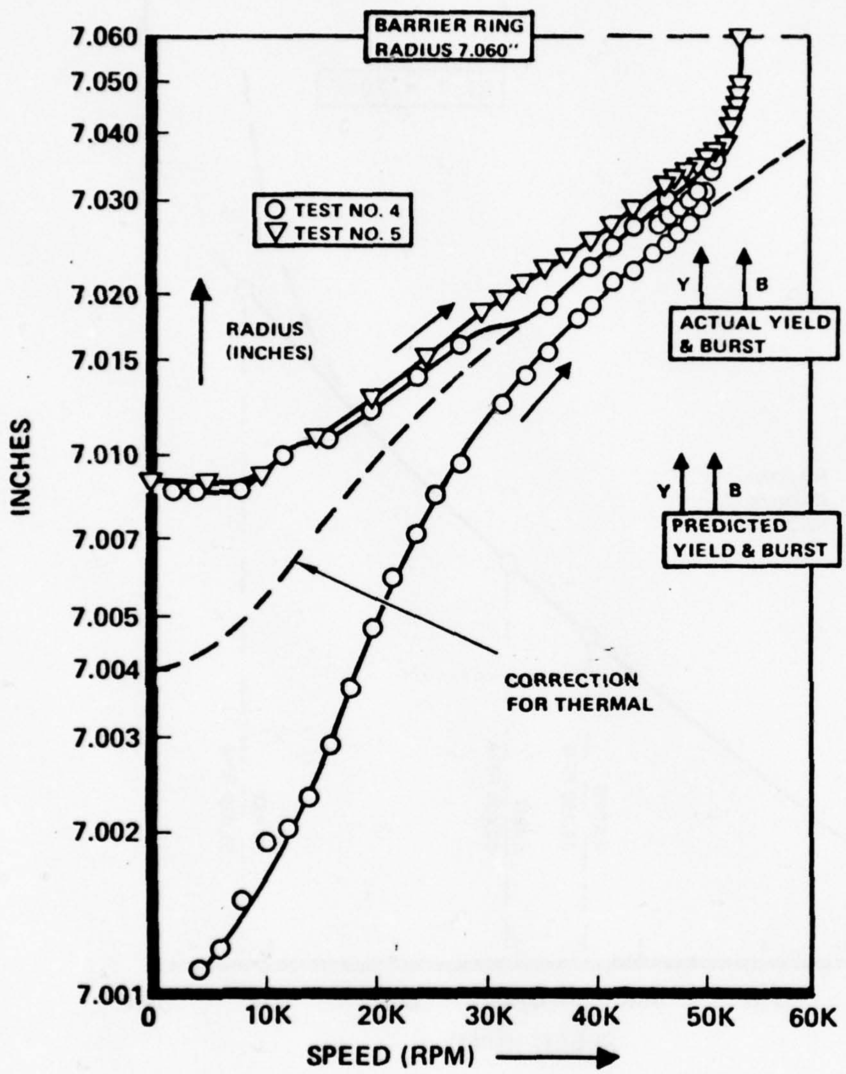

Figure 23. Subscale Rotor Experimental Radius vs Speed Data

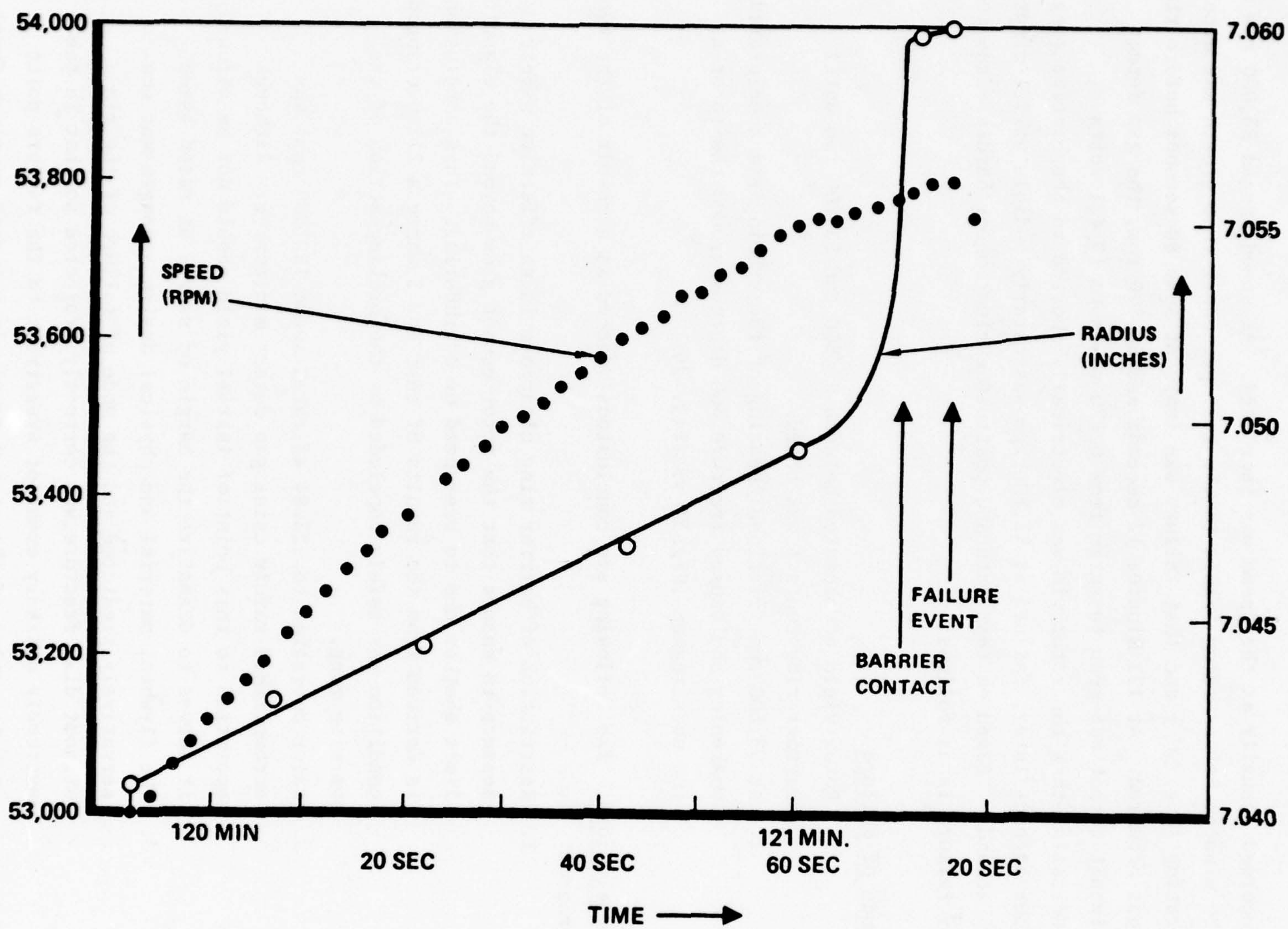

Figure 24. Rotor Test No. 5, Speed and Radius vs Time

recorded manually as the speed was increased. As speed passed 53,000 rpm at 120 minutes it was recognizable to the test personnel that growth was accelerating (Fig. 24) and that failure was imminent some 80 seconds before rim loss occurred. At 121 minutes 12 seconds and 53,790 rpm, the gap detect circuit output was seen to surge from 6.870 volts to 13.421 volts indicating that the rotor rim was electrically shorted to the barrier ring. Four seconds later, failure at 53,800 rpm was clearly audible at 121 minutes 16 seconds. Based on test data and posttest evaluation of hardware the mode of failure is as follows:

Mode of Failure

1. Rotor yield accelerated between 52,500 and 53,800 rpm until barrier ring contact was noted.
2. At 53,800 rpm frictional heating of the rubbing rim accelerated weakening and caused fracture 360 degrees around the rotor at the neck region (Fig. 25 vs. Fig. 26)

Conclusions. The following are conclusions reached as a result of the test program:

1. Restriction of barrier ring clearance is an effective safety measure to ensure that the rotor cannot grow beyond the elastic limit whether due to overspeed or overheating. This conclusion is derived from the results of test No. 3 where a 21% overspeed condition was safely concluded by the braking action of the barrier ring.
2. Rotor operation to 125.8% of rated speed (50,000 rpm) was accomplished safely using gap detect monitoring. Although operation to this point of initial yield should not be allowed, it serves to dramatize the margin of safety at rated speed.
3. The flywheel material and physical design strength was conservatively predicted and the mode of failure of the rim without disk fracture was correctly projected so that in the extremely unlikely event of overstress to the failure point only a portion of the stored energy is released as centrifugal momentum of fragments.

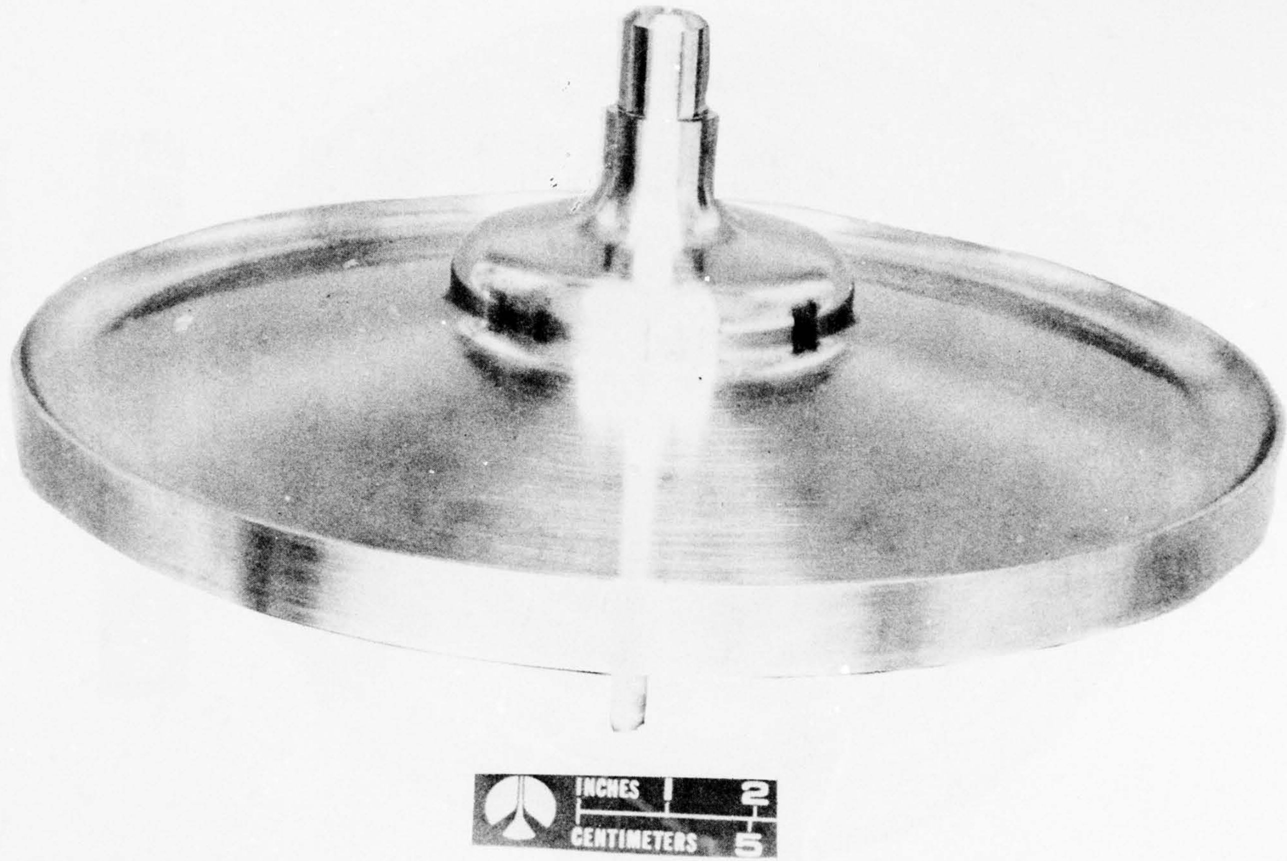

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Figure 25. Subscale Rotor - Pretest

4EF36-5/28/76-C1B

Figure 26. Subscale Rotor-Posttest

Accomplishments

1. The highest speed of operation without permanent deformation was identified during test #4 as 49,500 RPM (24.5% over rated speed.)
2. During test #4 it was indicated that permanent deformation in amounts as small as .001 to .002 can be detected during rotor operation.
3. The capability of monitoring rotor operation at 95-96% of burst speed was demonstrated during test #4 as speed was safely varied from 0 to 50,520 RPM, reduced to 45,460 RPM, returned to 51,520 RPM and reduced to 0 RPM without loss of integrity.
4. Test #5 served to verify the precision of stress analysis employed in the design of the flywheel rotor wherein failure of the rim was planned and demonstrated without incurring loss in integrity in the hub section.
5. Progressive disc growth to failure was monitored and recorded during test #5 providing a parametric verification of the stress/strain curve for permanent deformation versus speed.

The subscale phase of this program also served to verify other important predictions as follows:

1. The selected material, HP 9-4-30, can be machined and EB welded in the fully heat treated state to obtain a product of high quality and predictable performance.
2. The anticipated material properties of HP 9-4-30 such as strength and elongation were demonstrated to be valid, qualifying this material for high performance flywheel design and construction.
3. In the case where conditions are allowed to proceed to burst, the flywheel failure mode results in less destructive energy than alternative configurations yet has a higher performance index in terms of stored energy per pound of material at any given stress level.

MOD. P00010 - Brake Assemblies

During July of 1977, analysis and preliminary design studies were begun to define a suitable device for normal and emergency declaration of the flywheel module rotors during test. It was concluded that MK 15 turbines (normally used to power the J-2 rocket engine fuel turbopump) could be adapted to satisfy the need. The MK-15 turbopump is a developed product rated at 8749 shp and driven by 732 PSIA LOX/LH_2 combustion gas at 1296°F and 7.6 lbs/sec. Although this power level would be desirable in the interest of rapid flywheel declaration, it was concluded that reduced power would be acceptable for the sake of simplicity and system reliability. Therefore, an existing test facility pressurized air system was selected to drive the brakes. In this RS-31 application the brakes are designed to rotate in a vacuum (to minimize parasitic windage losses) and in reverse when driven by the flywheel rotors. For braking, the turbine exhaust is vented to ambient before 135 PSIA air at 60F and 7.0 lbs/sec is admitted to each brake inlet manifold. At 15,000 RPM the air flow is predicted to produce a braking torque of 1362 ft-lbs at each flywheel interface. Torque decays as flywheel speed is reduced resulting in a net time to stop of 117 seconds.

Since the flywheels are contra-rotating one brake must be mounted facing aft and exhausting toward the flywheel (left side) and the other brake faces forward and exhausts away from the flywheel module. Thus the interfacing structure, seal and bearing designs for each brake are distinct. During late 1976 and early 1977 design and analysis of the two brake configurations proceeded with considerable design inovation required to avoid modes of resonance associated with the several variables of operation and installation (forward driven, aft driven, freewheeling and powered). On 14 June 1977, authorization was received to initiate procurement and fabrication of the several new interfacing and housing components as well as modification of the MK15 turbine parts as required.

Two GFP turbopumps were disassembled and the gas generators (an integrally welded part of the turbine manifold) were cutoff with cover plates welded

over the opening. All new components were completed by late 1977, however, assembly of the two brakes was deferred until April of 1978 to give priority to assembly of the flywheel module.

PHASE II DETAIL DESIGN

Flywheel Rotor Assemblies

Two independent flywheel rotors are housed within the RS-31 case assembly. Each rotor consists of two 37.1-inch diameter discs which are bolted together and bolted to end hubs. Each 1495 pound rotor assembly has a total inertia of 159,486 in^2 pounds so that the pair of rotors store 30 KWH in the RS-31 module. at 14,506 RPM (or 32 KWH at 15,000 RPM). The basic rotor configuration was defined by phase I tradeoff studies and phase II activity was limited to optimization and detail design.

At the outset of phase II, assembly of the four rotor components was designed to be accomplished by performing 3 electron beam interface welds. Subscale rotor EB weld assembly had been accomplished with no problems according to procedures which involved demagnitization of the components, preheat of the weld area, weld and post-heat for relaxation of residual stresses. When it was determined that recurring problems with larger scale weld equipment and assemblies (comparable in size to the full-scale flywheel) were occurring, these potential risks, combined with the more difficult process of pre and post heating the larger disc weld areas, led to development of a new design for the assembly process. This final design accomplishes integration of the four component parts by use of bolts and pins at each interface. All bolt holes are thru flange sections of the components. The two discs are joined using 32 studs which have a 3/8" dia portion which is inserted into one disc flange and locked by slotted-thread key. Alignment pins are also placed into that disc flange. These pins and a pilot circle of the disc serve to align and rigidize the mating discs. The second disc is lowered onto the circle of studs and pins and lock nuts are used at the protruding 5/16" dia end of the studs. A series of bend-tab plates are used under the lock nuts to blank off pin axial freedom and to further lock the nuts down. At each end hub bolts are inserted into the threaded disc flanges to hold the hubs in place. Alignment pins and lock-tab plates are used here again and the parts are also piloted at their enter edge. As assembled the three mechanical interfaces are restrained from rotational slip by face friction with safety

margins of 1.88, 6.98 and 5.35 (FWD/MIDDLE/AFT) and in addition are restrained by pin shear resistance with independent safety margins of 3.40, 9.12 and 9.65 respectively.

The forward end of the rotor ends in a 2.2" pitch dia male involute attachment of the electric generator using a splined shaft to carry torque. The spline relief area with a diameter of 1.989 inches is the smallest output torque carrying station of the shaft resulting in a torsinal stress safety factor of at least 9.43.

The bearing journalsat each end of the rotor have a common diameter of 3.1499 inches providing a press fit for the bearing inner races A pair of 3/16" dia holes are drilled radially in the aft hub shaft to allow lube oil to enter an axial passage in the shaft for transmission aft to clutch feed orifices. The clutch assembly is mounted over a double row of 2.0 pitch diameter male involute splines near the aft end of the aft hub shaft. The rotor discs are thickest at their center gradually tapering to a minimum thickness at the neck and then expanding to the rim.

The flange circle for the disc-hub interface has been designed so as to minimize the joint stresses which arise from non-equal strain of the mated parts at elevated speeds. Furthermore this flange circle is selected for minimum participation in the disc stress distribution so that the combined interface stresses remain less than the disc's base or neck peak stress values.

In addition to the various bearings, spacers, clutches, etc. which are mounted on the shaft, as discussed later, each shaft is fitted with a speed sensor wheel near its aft end. This eight tooth wheel acts in conjunction with an electrical coil sensor to provide 8 pulses per revolution or 2000 Hz at 15,000 RPM.

CASE ASSEMBLY DESIGN

The RS-31 case assembly is a 45.5-inch long axially bolted stack of three relatively shallow flanged sections, each of which are 5 to 6 feet wide and about 4' tall. All three cases are machined from sand cast 356-T6 aluminum alloy. (Figure 27)

During the phase I design study, Tens 50 aluminum alloy was selected for case fabrication, however, it was subsequently replaced by 356-T6 aluminum to reduce cost (by using a commercial rather than aerospace grade material and foundary). The nominal ultimate and yield tensile properties of 356-T6 (at 30 KSI and 20 KSI) are approximately 25% below those of Tens 50, however, it was concluded that the thicker sections and heavier weight of the 356-T6 castings would be preferable to the more expensive Tens 50 castings.

In general, the as-cast 356-T6 structure walls are about 1" thick resulting in total weight for the three machined cases of 2021 lbs.

The forward case acts as the interface which supports the two Bendix electrical generators, provides a cylindrical bore for each forward bearing and seal assembly and houses the right hand flywheel rotor assembly and its mating barrier ring set. Its axial thickness is about 20 inches.

The center case serves as a closure plate and rear bearing cavity for the aft end of the right hand rotor and houses the left hand rotor and its mating barrier ring set. Its axial thickness is about 14-inches. The aft case serves as a closure plate and rear bearing cavity for the aft end of the left hand rotor, contains an oil collection chamber and an 8.35 gallon oil sump and provides the support interface for the gearbox and engine. The aft case is about 12" thick axially.

Epoxy glass laminate rings are bonded into the interior case walls and machined to provide pilot surfaces for support of the four barrier rings. Steel plugs

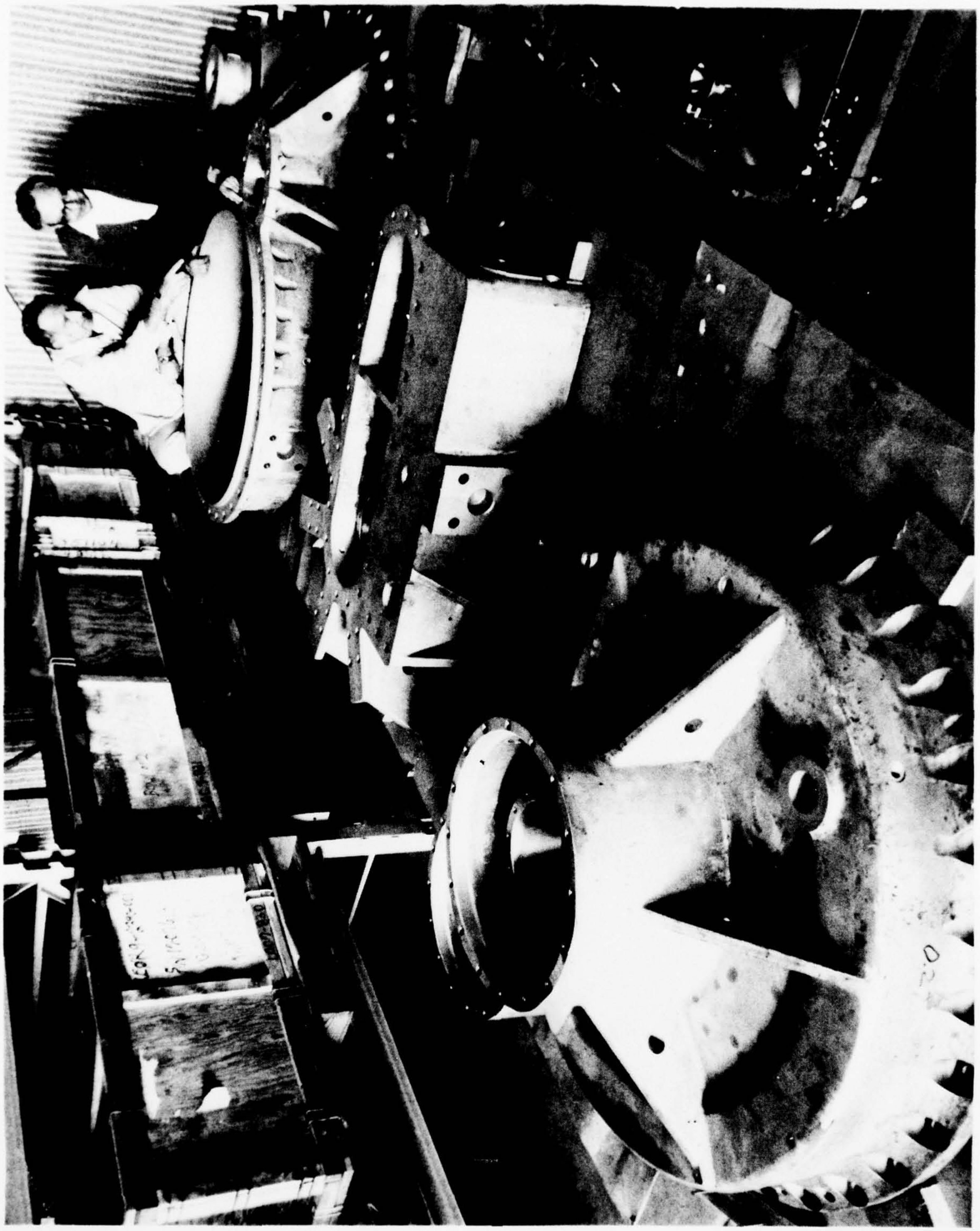

Figure 27. RS-31 Case Sections

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are bonded into the case walls in eight locations to serve as platforms for accellerometers. Two inch pipe-tap holes are provided at four locations to accept visual observation windows. O-ring grooves are machined into each rotor chamber interface. A drain port from the oil collection reservoir is designed to accept a 6.5-inch diameter conical de-aerator screen assembly. The oil sump ports are designed to accept installation of a 1.5 KW thermo stat immersion heater, a liquid level float switch and a temperature control valve thermowell. Oil flow passages are drilled into the case walls to lubricate and drain each of the four bearing and seal assemblies and ports are provided at the bottom of each rotor chamber for moisture collection, drainage and case evacuation. Out of consideration for the overhang accessory moments at each end and the large diameter shallow depth vacuum chamber structural load, radial webs are included in the as-cast structure to provide rigidity to the case walls and the overall case assembly.

Each case has a single base pad resulting in three point support for the assembled module.

Gearbox Design

Preliminary design of the gearbox at Rocketdyne was limited to definition of the drive train and casing configuration, envelope and interface as required to satisfy the system criteria. This preliminary work allowed module design to proceed based on a 28" rotor axis separation, proper directions of rotation, clutch envelopes and spline sizes, drive shaft lengths, weights and moments for dynamic analysis and case structural form required to support the torque and dead weight moments. The gearbox detail design and fabrication responsibilities were subcontracted to Barber-Nichols Engineering Company of Arvada, Colorado in July of 1976. As detail characteristics of the gearbox design became available each element of the design was subjected to intensive review and analysis by appropriate specialists at Rocketdyne to assure that all aspects of the design would perform as specified. These studies included aspects such as lubrication adequacy, rotor dynamics, bearing tolerances, scoring analysis using Maag and Dudley scoring indices, Kelley Blake Agma Flash Temperature, Bodensieck film thickness criterion, reverse bending cycle life, crowning and alignment tolerances.

The gearbox casing, which is suspended from a webbed face of the module case, is machined from an Allmag 35 aluminum alloy casting whose ultimate and yield tensile strengths are 41 KSI and 21 KSI respectively. Elongation is 16%. This material was chosen for its exceptional freedom from locked-up stresses and its dimensional stability allowing gears and bearings to be piloted directly into the heat-treated and finish-machined casting. The gearbox case also supports the 750# driver engine which is suspended from a bolt circle at the left rear gear axis. Bearing pockets are machined into the rear internal face of the case in four places and a machined aluminum enclosure plate supports the second forward-end bearing for each of the four power gears. Each 72 tooth end gear has a pitch diameter of 7.2 inches and the two 104 tooth idler gears have a pitch diameter of 10.4". All four 9310H steel gears have a face width of 2.38", are case hardened to 58 to 63 Rc and are crowned 0.0005". The left end gear is internally splined to receive the drive shaft of the engine and the output end of the drive shaft to the left hand clutch. The right end gear

is also splined internally to drive the rear face mounted gearbox oil pump and the forward output shaft to the right clutch. The clutch shafts have male splines at the gear end and female splines at the clutch end and the clutch assemblies are recessed within the female socket. All three shafts and both clutches were included under Barber-Nichols sub-contract. The sprag-type clutch cages are held between two concentric cylinders with 113H Barden ball bearing assemblies at each end of each assembly. The inner clutch cylinder is female splined and pilots to the flywheel rotor assembly aft hub shaft. The outer clutch cylinder is male splined and freely inserted within the drive shaft socket and is crowned in two axiis for maximum acceptance of misalignment.

The gearbox lube oil pump is a Tuthill gear rotor unit producing 9 GPM and 40 PSID at 15,000 RPM. A 10 micron oil filter is included upstream of the pump and discharge flow is distributed internally to three mesh jets and four bearing jets at the case plate by steel tubing. Bearing flow sprays upon the forward bearing and flows thru the two hollow idler gears and discharges thru the idler rear bearings to the sump.

The end gears which are filled internally by drive shaft bulk are provided with extra lube jets at the aft face to lubricate the aft bearings. The left splined clutch shaft is drilled to receive oil flow from the left flywheel rotor for lubrication of its gear spline face and the engine drive shaft is also drilled so that this flow can continue aft to lubricate the engine drive socket. Oil flow to the engine is able to drain back into the module thru two holes at the bottom of the gearbox/engine interface.

Upon completion of critical design review by Rocketdyne in September of 1976 at Barber-Nichols, authorization was given to proceed with release and fabrication of the gearbox subsystem.

Other Details of Design

Rotor Seals. Four rotor shaft seals are used in the assembled module to permit each separate rotor chamber to be held at an evacuated pressure level of about

0.1 PSIA. The outboard face of each seal is exposed to a bearing oil scavenge zone at a nominal pressure of 14.7 PSIA. Out of consideration for limiting the vacuum pumping capacity to a reasonable value, 5 SCIM was defined as the maximum allowable air leakage criteria for each seal. Since the vacuum pump serves both chambers the total allowed leak of 20 SCIM converts to 1.7 ft^3/min pumping capacity at 0.1 PSIA, well within the 4.5 CFM capability of the selected pump allowing some margin for oil vapor entrainment and static seal leakages.

The selected seal design is based upon many years of successful Rocketdyne experience with carbon nose spring loaded shaft seals. The challenge in this particular application was to design a seal whose face load was very low without losing freedom of axial motion to follow shaft movements. Low face load is desired to minimize parasitic losses and to avoid heating and distortion at high contact speeds.

Spring load is supplemented by a pressure/area closing force. With minimum closing force as low as 11.33 lbs, friction of the encapsulated seal assembly O-ring and the spring/case interface must be minimized.

The carbon nose of the seal assembly interfaces with a 4130 steel mating ring whose surface is chromium plated and lapped to a precision mirror finish. The mating ring is locked onto and spins with the rotor resulting in an interface seal rubbing speed of about 273 ft/sec. Under nominal conditions of full delta pressure and full face contact, single seal sliding friction is estimated to produce only about 0.09 BTU/sec, well within the heat dissipation capability of the design.

Rotor Bearing System. Each flywheel rotor is positioned axially by a duplex pair of opposed angular contact 80 mm bore, 125 mm O.D., ball bearings located at the forward end of the module case assembly. These bearings are supported within a cylindrical 4340 steel holder which is rigidly fixed at the far forward end, but radially free at the bearing station. Radial deflection is controlled by the beam stiffness of the holder and damped by an oil annulus between the bearing holder and a secondary 4340 steel cylinder which lines the aluminum case bore. Stiffness of the bearing holder is precisely controlled by a

pattern of windows in the holder wall between its support station and the bearing station. This design is prescribed to reduce the stiffness of the bearing system for control of resonant frequencies and to damp oscillation as the rotor speed passes thru the first critical speed and approaches the second critical speed. The bearings are designed for a press fit at the bore and a loose fit at their OD.

The forward bearing system lubrication supply delivers 1 GPM at 60 PSI to each forward bearing pair thru six jets (three per bearing). Oil flows from between the bearing pair forward and aft thru the bearings and drains into the case where it is scavenged for return to the supply system. Under nominal conditions at rated speed bearing viscous friction will produce enough heat to raise lubricant temperature about 50°F assuming a 200°F local operating temperature and 23% of the heat dissipated thru the case structure.

The aft end of each rotor assembly is supported by a single 80 mm bore, 125 mm OD, cylindrical roller bearing. These aft bearings are also supported in a spring-damp system similar to the forward bearings with bore press fit and loose OD fit. The aft bearing lubrication is extracted from 500 psi flow to the rotor shaft clutch bearings and drive splines so that flow to the flywheel bearing is increased to a maximum of 1 GPM per bearing at limit rotor speed when shaft centrifugal rejection is at its peak. At zero RPM roller bearing oil flow is 0.37 GPM thus the supply versus speed curve is proportioned to the lubrication needs of the bearing allowing 91% of the heat from bearing friction to be carried away with a 50°F oil temperature rise assuming only 9% dissipation thru the case.

The aft end bearings are designed to accommodate rotor axial length variation-initially due to -0.028" poisson shrinkage and later due to + 0.026" growth.

Barrier Ring Design. The RS-31 barrier ring installation consists of a set of four 37.242" I.D. rings - one surrounding each of the four 37.100" diameter flywheel discs. These rings serve as overspeed breaking devices and fracture

AD-A060 351 ROCKWELL INTERNATIONAL CANOGA PARK CALIF ROCKETDYNE DIV F/G 10/3
DEVELOPMENT OF A HIGH ENERGY STORAGE FLYWHEEL MODULE.(U)
MAY 78 D R HODSON DAAG53-75-C-0278
UNCLASSIFIED RI/RD78-207 NL

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fragment barrier devices as well as acting as a part of the rotor growth sensing circuit. They are not intended as total containment barriers. Design for total containment of failure would require a significantly larger housing diameter with provision of additional void space around each ring to allow for elastic expansion of the ring. As built the ring is designed to withstand the load of rim failure and direct centrifugal penetration by fragments. Prevention of failure, however, is the primary role of this barrier ring installation. Fig.7 describes rotor growth as a function of speed where it will be noted that rotor material yield begins at 17,600 RPM and burst is predicted at 19,200 RPM. By sizing the barrier ring to be 0.071 inches larger in radius than the rotor, interference and braking will begin to occur at 16,500 RPM.

Each ring is supported and piloted upon an epoxy glass laminate ring bonded and machined into the housing. A set of two axial guide rings located between each pair of barrier rings are also insulated on their interface surfaces so that the barrier rings are electrically insulated from the module. The guide rings are spring loaded apart and have three steel ball anti-rotation pin stations. Insulated electrical posts thru the case wall are spring loaded into each barrier ring to provide non-stressing contacts for gap sensing circuit signal conductance.

Control Systems Design

Lubrication and Evacuation Equipment. An 8.35 gal lube oil sump has been designed into the rear case of the flywheel module. Sump level is detected by a liquid level switch at port LO-3 in the sump activating an indication panel lite when sump level is below 5 gallons. Oil in the sump is heated to 150°F by a locally adjustable calrod heater at port LO-1. Oil flow to seals, gears and bearings is drawn out of 1" port LO-4 where it passes thru a magnetic particle separator and then a 40 x 40 mesh strainer. Oil to the gearbox splits off at this point thru a 1 1/2" flapper check valve used to hold a prime for the elevated gearbox driven positive displacement gear-rotor pump. A 10 micron filter is located between the check valve and this gearbox

pump. At full gearbox speed, flow thru this branch is 9 GPM at 40 PSIG serving three gear mesh jets and eight gear bearings. Gearbox oil is exhausted into a rear module case chamber above the sump and scavenged as discussed later. A second branch of the oil supply line is drawn thru a 60 PSIG electric motor driven gear-rotor pump at 2 GPM. This flow proceeds thru a 10 micron filter to feed the two forward flywheel rotor bearing and seal sets thru ports LO-9 and LO-10. The forward cavities are both scavenged (thru ports LO-11 and LO-12) by a small electric-motor-driven pump which returns the oil to the aft case upper chamber thru port LO-14. A third branch of the oil supply system passes thru two parallel 500 PSIG electric motor driven gear pumps at a total of 7.5 GPM and thru a single 10 micron filter to feed all rear end elements thru ports LO-7 and LO-8. Rear flow is distributed between the two main rear flywheel shaft bearings, the 4 clutch bearings, the shaft seals, and several shaft spline interfaces. All of this third branch flow falls into the upper chamber of the rear case. Due to the variable nature of the gearbox flow demand, total flow thru the system varies from a 9.5 to 18.5 GPM. All of this flow drains out of the rear case upper chamber thru a de-aerator screen within the case and is drawn out of the case thru a 40 x 40 mesh strainer by a large gear-rotor scavenge pump. If the sump oil is cool (below 15o°F) it is allowed to transit directly back to the sump thru a 1" temperature control valve, however, because of pre-heating and heat picked up in transit the TRV is normally closed forcing scavenged oil thru an air fan cooled heat exchanger before return to the sump. As cooled oil influences sump oil temperature the TRV begins to open to achieve the desired 150°F mix back to the sump. A 15 PSIG pressure regulating value between the cooler and the sump serves to restrict flow thru the cooler as the TRV opens.

Evacuation of the flywheel module's two separate rotor chamber is accomplished by a 150 liter/minute motor driven vacuum pump which draws from ports VAC-5A and VAC-6A. Solenoid vacuum values in each suction line may be remotely opened or closed to check vacuum retention during operation.

Most of the oil and vacuum system equipment is located on one of three packages (Figure 28, 29) adjacent to the module. Pumping assembly LE76-032-ER includes a 5.7 hp, 208V, 3ϕ, 400 Hz motor which drives the three oil delivery pumps with relief values and line filters. Pumping assembly LE76-034-ER includes a second identical motor which drives the two scavenge pumps and the vacuum pump and includes a strainer and one relief value. The cooler assembly is a free standing assembly with two 1.0 hp, 208V, 3ϕ, 400 Hz, electric motor driven fans rated at 1000 CFM.

The RS-31 lubrication and evacuation systems are designed to be integrated with backup facility systems for development-test. Sheet 2 of drawing LE76-071-ER describes facility criteria for backup systems as well as for operation of the air driven brake assemblies used during test of the module.

<u>Electrical Equipment</u>. The RS-31 electrical systems are described by drawing LE76-067-ER. Supply power is 208V, 3ϕ, 400 Hz which is used to power the four electric motors and is connected so as to derive 120V, 1ϕ,400 Hz power for main switches, relays DC power supplies, vacuum instruments control modules and the oil heater unit.

<u>BREAKER PANEL</u> - Main power is initially supplied to a panel where three 30A amp circuit breakers protect each pump motor and three 5 amp circuit breakers protect each fan motor in the 3ϕ circuit. Single phase power to the oil heater is directed thru a single 20A circuit breaker. A set of four relays on the breaker panel allow for remote command of three phase power delivery to the oil pump and fan motors.

Heater power delivery is controlled automatically by a dial set thermostat at the element. Single phase power is delivered from the breaker panel to the controller panel.

<u>CONTROLLER PANEL</u> - A switch on the controller panel (drawing LE76-070-ER) directs single phase 120V, 400 Hz power to all controls on the panel thru three 5A fuses. Fuse F1 serves all switch and relay circuits while fuses

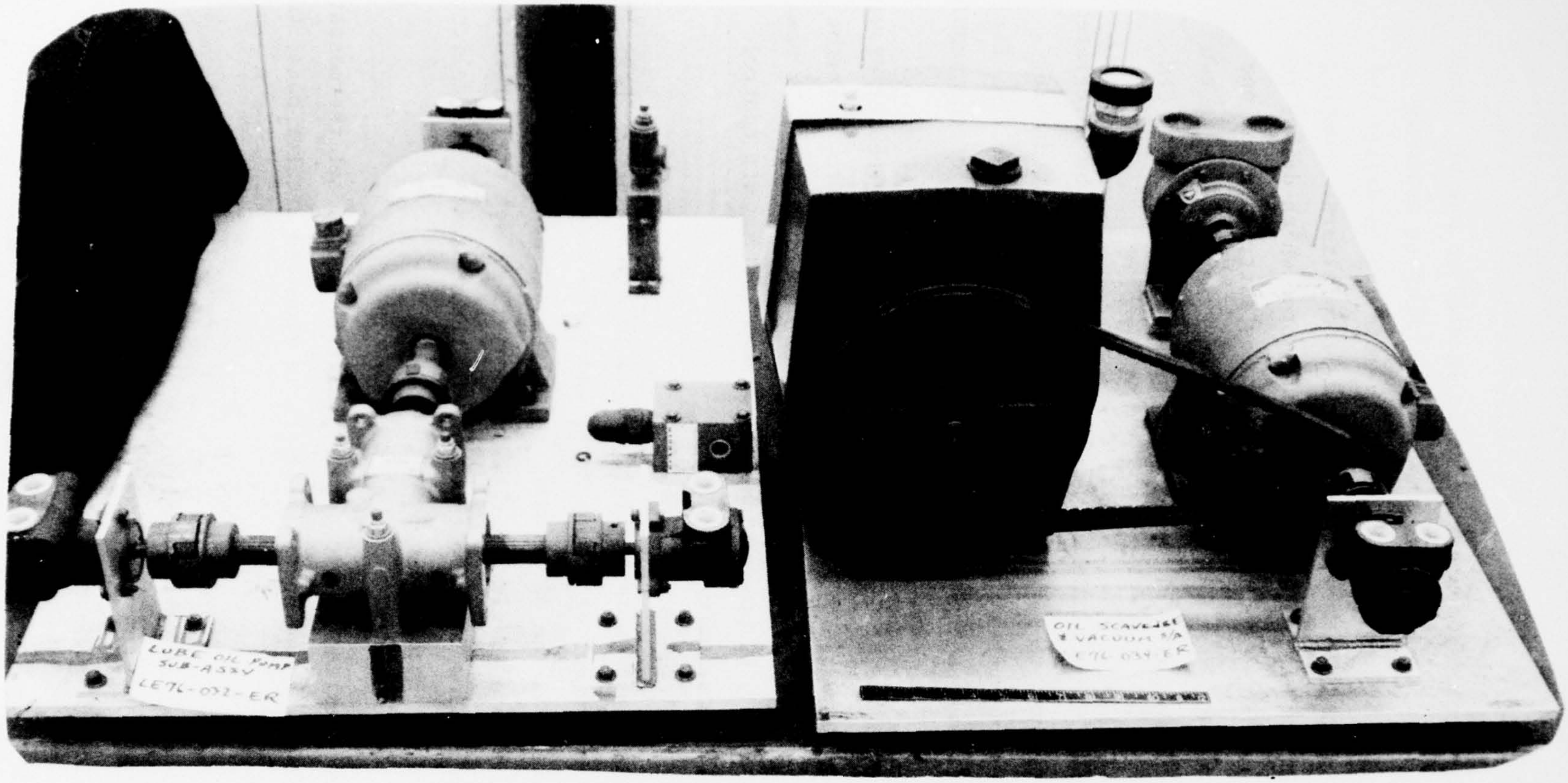

Figure 28. Lube Oil and Vacuum Pumping Assemblies

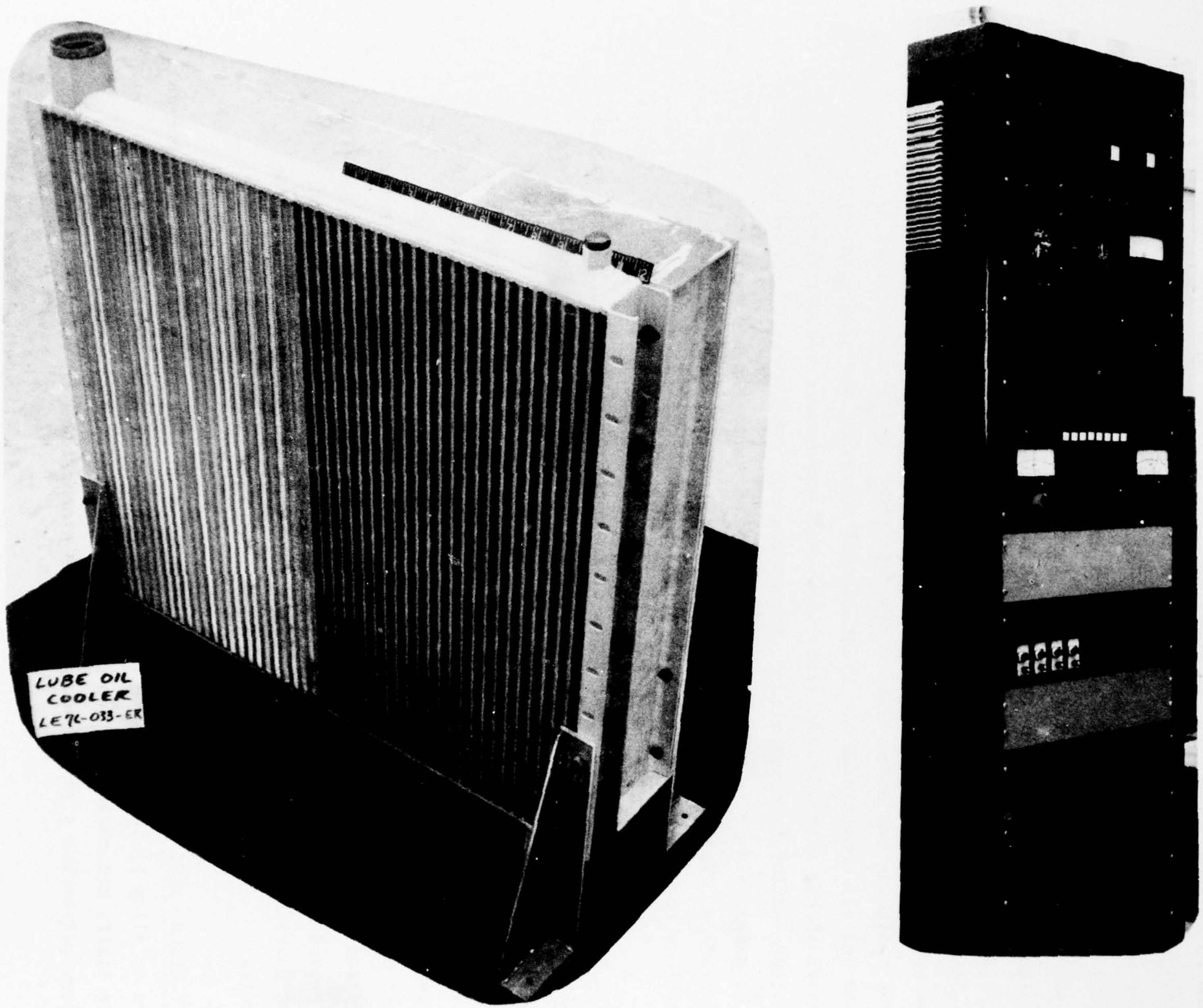

Figure 29. Oil Cooler and System Controller Assembly

F2 and F3 serve a set of four 2A power supplies which produce 5VDC power for digital indicators and alarm circuits. A set of eight switches in the lower center of the controller panel are provided to operate the vacuum valves, the oil and vacuum pumps, the cooler fans and the brakes. Three of these switches are also used to perform other functions such as reset alarm circuits, select measurement channels and to lockout alarm signals to avoid trip. Operator trip is accomplished using a switch in the upper right corner of the panel opposite the power switch. A set of 10 alarm lites between the power and trip switches serve to notify the operator of status including run mode, alarm, trip, oil level, oil pressure, loss of vacuum and oil contamination. Eight digital meters on the panel indicate, flywheel speed, rotor growth, vibration, case temperature and oil temperature. Two other indicators display case vacuum pressure. A 5 position rotary switch on the panel is used to switch instrumentation between run and calibrate modes of operation. Relays behind the panel are used to allow run, alarm and trip conditions to be remotely activated or monitored.

SIGNAL CONDITIONING PANEL - Two types of instrumentation require special signal conditioning as provided on this panel per drawing LE76-069-ER. Four Endevco Model 2721 charge amplifiers provide signal to accelerometers on the module and proportional DC signals to the data system. Four Rocketdyne developed modules in the form of printed circuit cards on the panel provide excitation to the flywheel rotor gap sensing surfaces and deliver a proportional DC signal to the data system. Because of the common ground of all the rotor discs and the high EM1/RFI field environment, each of the gap detect circuits are provided with a separate isolated 15VDC power supply module. A fifth 15VDC power supply module on this panel powers all four vibration detection circuits.

LIMIT-ALARM PANEL - A set of 18 plug-in modules on this panel (drawing LE76-068-ER) serve to allow hi-low limits to be set for monitor of data signals and alarms where limits are exceeded. Two of the modules are speed signal conditioning devices converting signal frequency to the proportional DC voltage. The other

16 modules have hi-low set pots and relays which activate when the set point is reached. In addition, each module includes a transmitter circuit which allows the input signal to be amplified, zeroed and spaned for transmission to indicators and recorders.

The GFP gas turbine engine is operated and monitored by a GFP operator control panel. Control of the flywheel module is limited to provision of lube oil vacuum conditions appropriate to powered operation, however, (for test purposes) additional controls are provided for two air powered braking turbines. Appendix A describes the complete operational procedure for startup and shutdown of the system including engine and brake control. System deliverable instrumented parameters consist of rotor and gearbox speed, bearing case and oil temperatures, case and oil pressures, rotor growth and vibration. The speed signals are transmitted thru limit-alarms which are set to notify the operator of low operating speed (below 10,500 RPM), high operating speed (above 15,000 RPM), and overspeed (16,000 RPM). At rated speed of 15,000 RPM rotor radial dimensions will grow reducing gap clearance by .048 inch. At 16,000 RPM the radial growth will be .055 inch at which point preset alarm signals are activated.

Figure 30 describes system control schematically.

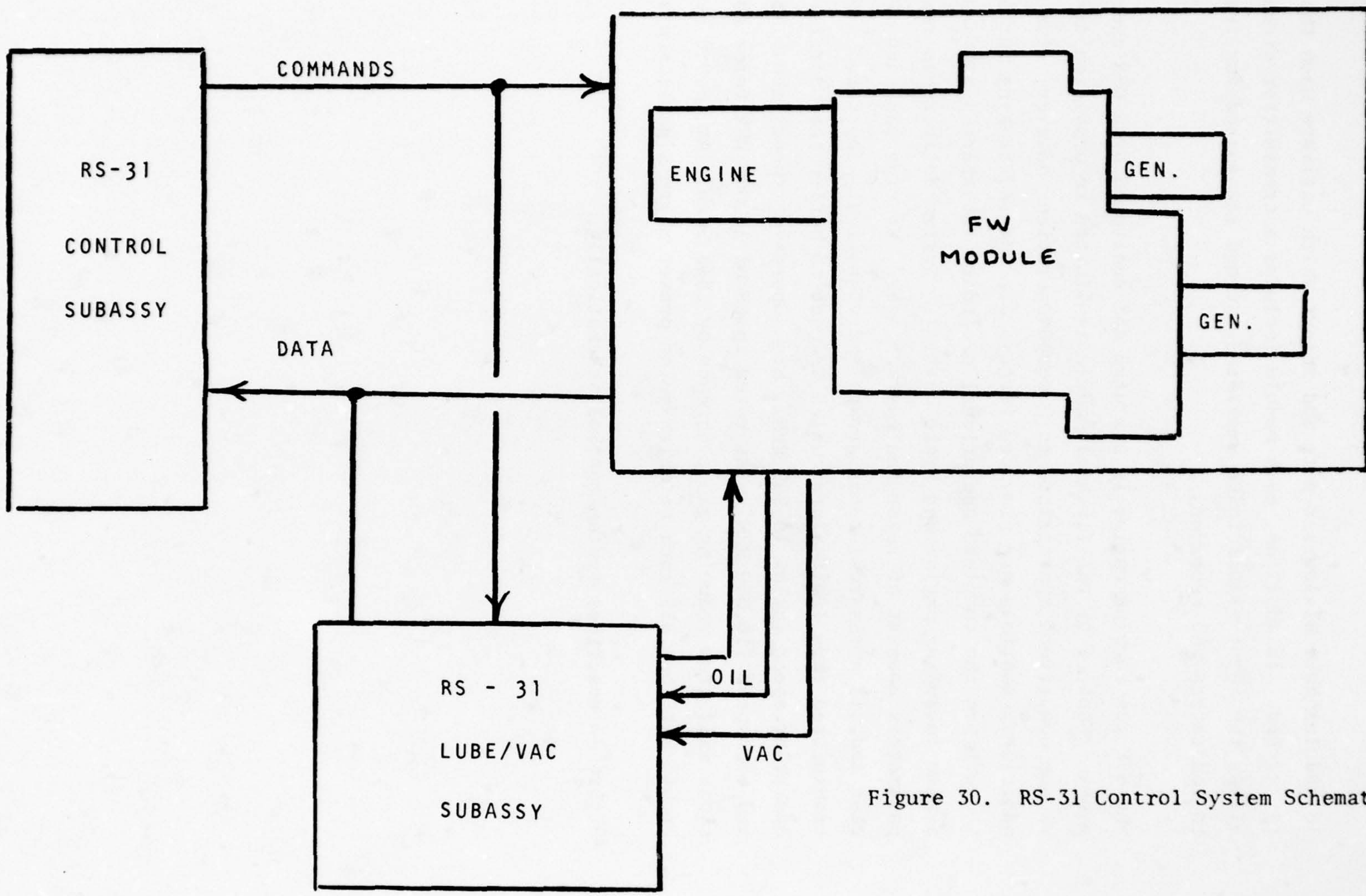

Figure 30. RS-31 Control System Schematic

PHASE III FABRICATION & ASSEMBLY

Fabrication of unique new hardware whose ultimate properties are critical to safety and performance of the end product requires that an extra degree of process control be employed. The requirements are beyond shop experience, habits are not properly formed and production jobs tend to receive priority management attention. Similarly inspection by product line quality control people can fail to adequately spot significant defects. For these reasons all flywheel rotor fabrication activities were monitored exclusively by the responsible Rocketdyne design and material processing engineers. Forging of the new discs and hubs, heat treat, ultrasonic, magnaflux and dimensional inspections, machining, assembly and balancing operations were each witnessed by a qualified engineering specialist. Similar attention to other components such as the housing casting process, barrier ring fabrication, controller assembly and module assembly was instrumental to first-hand confidence in the quality of the work.

Rotor Assemblies

Rotor disc and hub blanks were drop-hammer forged using certified vacuum melt HP 9-4-30 material beginning in July of 1975. Five discs were forged and the fifth disc was processed thru heat treat for cut up and analysis of section properties. Concurrently, a single, long L/D, cylindrical forging was made from which five hub shafts were cut and the fifth hub was analyzed. Following acceptance of the sample test properties, the remaining disc & hub forgings were roughly cut to within about 1/4 inch of their final overall envelope dimensions for delivery to Rocketdyne.

After the rotor assembly design was optimized in January 1976, these forgings were rough machined to within about .060" of their detail contours and submitted for heat treatment consisting of normalize, austenitize, refrigeration and tempering cycles. The hubs were treated first and upon subsequent inspection it was noted that three of the four hubs had suffered cracks at the corners where the shaft transitions to the rotor attach flange plate.

This anomaly was reviewed with the Republic Steel Co's metalurgists, with other material process engineers who had experience with HP 9-4-30 and with the forging and heat treat house metalurgists. No single aspect of the process or the hardware was found to be inconsistent with accepted industry practices, however, in the search for reduction of risk for subsequent trials, two process changes were made. Radii of the hardware were increased and a more conservative quench process was imposed. Replacement hub forgings were then prepared and a sample hub and test disc were subjected to heat treat with thermal instrumentation. From these test data, adequate confidence was derived and heat treat of the original discs and the new hub forgings proceeded with complete success.

Final machining of the rotor hardware was initiated in October, 1977 with extreme care given to checkout and operational precision of the tracer lathe installation, the shaft spline process and interface surface grind operations. A very small but significant hysteresis was noted in the tracer lathe hydraulic valve function which required replacement of the valve and test runs on sample material before allowing disc surface contours to be cut. Sample splines were cut and inspected using precise Rocketdyne inspection tools. Imperfections were found and corrective action was taken to adjust the supplier's spline tooling appropriately. Although it was anticipated that normal machining procedures would achieve the specified interface precision, an agreement was ultimately reached to optimize prospects for success by going to a precision grinding shop for pin hole, pilot and surface fits. The advantages of that judgement were recognized when the rotor assembly process was begun. What had been expected to be a complicated and tediously detailed process (mating of the large interlocking, press-fit, pilot diameters), proved, in fact, to be easily and quickly accomplished because the fits were exact and the surface finishes were excellent.

Following journal grind of the assembled rotor, preparations for dynamic balancing were completed. The rotor assemblies were each prepared for balance by adding all of the shaft mounted components including the clutch assembly, the speed sensor wheel, the forward ball bearing assemblies, the aft roller bearing inner race and miscellaneous spacers, washers and nuts. At the forward end, the stationary

bearing support cage was also provided to assure that the duplex ball pair would be properly preloaded. At the aft end, a special roller cradle was fabricated to guide the roller bearing inner race. The aft rollers and outer race were not included because of their larger diametral clearance and consequent spin noise contribution. Each of the 9.75 lb. clutch assembly components were prebalanced during their fabrication to avoid adding significant inseparable unbalance outboard of the rotor bearings. The rotor balance assembly was then balanced in each rotor plane with final correction taken along the inside periphery of the disc's rim at a radius of 18.30 to 18.40 inches by skimming parent material from the sharp edge. Equal amounts of material were taken from each face of the selected disc to preserve disc symmetry about its radial axis. Total residual unbalance of the completed rotors was brought to within the allowed 15 gm-in criteria.

Case Fabrication

Fabrication of the three main module casings began with the design and fabrication of tooling. In consideration of the large size and the required module quality, it was anticipated that successful first-try casting of each of the three different case assembly sections would depend, for the most part, on the design experience and judgment of the pattern maker. Since hardware cost was very much a factor the ultimate selection of suppliers was arranged to utilize a very experienced aerospace quality pattern maker combined with a commercial-grade casting supplier.

With Rocketdyne's most experienced casting specialist's assistance and advice, the three large sets of wooden patterns were designed and fabricated during the period of October 1976 thru January 1977 and reviewed with the casting house for assurance that liquid metal flow and cooling would be properly controlled in the final sand core boxes which are produced from the patterns. In particular, care was taken to assure that the tooling would produce castings that would be free of residual stresses (which could lead to warping and tolerances outside of the envelope needed for machining). A second problem was to be certain that the material quality was high and free of defects in structurally critical areas and along surfaces which would ultimately need high surface finishes for the 43 inch diameter o-ring sealed interfaces of the two vacuum chambers. In certain other areas which are recessed deeply and nearly inaccessible with the oil

distribution and oil reservoir system, loose flashing could be a serious contamination problem.

Pour of castings began in March 1977. The first pour was unacceptable because of a foundry error but thereafter all three castings were completed without difficulty. During April and May, the raw castings were prepared for machining by removal of external risers, weld closure of core support voids, heat treat of the 356 aluminum to the T6 condition, x-ray inspection and dimensional inspection.

Final detail machining of the three cases began in June 1977. Initially such machining was limited to non-interfacing surfaces. Then the cases were subjected to pressurized impregnations using a polyester resin to fill the casting wall voids. Finally, the cases were anodized. Because of the thermal transients encountered during impregnation, seal interfaces, pilot surfaces and the rotor axis hoses were machined after impregnation and anodize were completed.

Gearbox Fabrication Assembly & Test

Release of authorization to Barber Nichols Engineering Co. to proceed with gearbox fabrication was constrained until completion of design review in September, 1976, but proceeded by procurement of long lead materials including casting patterns. Patterns were designed & prepared by C&J Pattern of Denver Colorado and successful case casting pour was accomplished by Artisan Casting Co. of Denver in November, 1976. Detail case and case details machining was completed by Bard-Newport of California in June, 1977. Gear and shaft work was also completed by Vistar Gear Co of Los Angeles in June 1977. Assembly of the gearbox was then initiated by Barber-Nichols in February following inspection and dynamic balancing of all shafts, gears and clutch parts. On February 17, 1977 a team of Rocketdyne engineering and purchasing personnel visited Barber Nichols for final inspection of the pre-assembly hardware. Following completion of assembly, bench testing was completed to check fit, static torque and backlash. During late March, 1977, no load rotational testing was initiated where test instrumentation identified a low flow condition at very high speed. This problem was traced to gear pump cavitation and was resolved by substitution of a higher performance subassembly. Subsequently, the gearbox assembly was subjected to 8 tests to rated speed of 15,000 rpm followed by 2 tests to 16,500 rpm for

5 to 10 minutes each. Although these tests were conducted in an unloaded mode, the assembly was subjected to conditions which were in excess of the intended application in terms of heat buildings in gears, bearings and case due to viscous and windage friction. Prior to the final (tenth) test the gearbox was diassembled, inspected for gear and bearing wear and found to be free of any signs of defect.

RS-31 System Assembly

Assembly of the module was initiated on 13 March 1977 beginning with placement of the aft case in a forward-face-up position on the work table and installation of an adjustable bearing (tool) in the left bearing cavity. The left hand flywheel rotor was lowered into this tool and adjusted, using the tool for proper axial position. After the aft left side barrier ring was set into it's case pilot around the aft disc of the rotor the middle case was lowered into place closing out the left rotor cavity. Before closure, the barrier separating ring installation was added and the forward left barrier ring was pre-positioned and locked into the middle case. A second dummy bearing tool was also used in the right rotor case bore which is located in the middle case. Next, the right hand rotor was lowered into the middle case tool and adjusted. Finally, the right side barrier ring stack was made and the forward case was added closing out the module.

With the rotors properly positioned final forward bearing and real installations were made locking the rotors into fixed axial positions. With the case assembly completed and bolted the assembly was lifted & rotated so as to rest on it's forward face providing full access to the aft bearing cavities from above.

At this point, the temporary bearings were removed and replaced with the final aft bearing and seal installation and the remainder of the rotating elements were added to the shafts (including speed wheel and clutch components). Final assembly operations included alignment and installation of the gearbox and engine at the aft end of the case assembly and addition of the MK-15 turbine brakes at the forward end of the module. Following completion of the case assembly,all parts were closed off and each individual rotor chamber was evacuated and held at 76 Torr for 30 minutes without any evidence of increase in pressure.

APPENDIX A

RC 1290

RS-31 TEST SPECIFICATION

PREPARED BY	CODE IDENT. NO.: 02602	NUMBER RC-1290
		TYPE
APPROVALS	**Rockwell International Corporation** **Rocketdyne Division** Canoga Park, California	DATE
		SUPERSEDES SPEC. DATED:
	SPECIFICATION	REV. LTR. A — PAGE 1 of 39

TITLE

RS-31 FLYWHEEL MODULE TESTING

1.0 SCOPE - This specification establishes the requirements for test evaluation of a pilot model dynamic energy storage system designated RS-31.

1.1 SYSTEM DESCRIPTION - The RS-31 system consists of an AVCO T55-L-7C gas turbine engine (GFP) driver mounted at the rear of a Rocketdyne developed module wherein two contra-rotating flywheel rotor assemblies are to be accelerated through a gearbox/clutch arrangement until the product of their inertia and speed constitute a level of kinetic energy storage suitable for delivery of short burst of high torque at the two forward end shafts when loaded by two Bendix (GFP) electric generators. The general system configuration is depicted by Fig. 7.1. In the test model, generators will not be used, however, a pair of Rocketdyne developed air powered turbine brakes (modified MK15F turbines) will be installed at the forward output faces of the module attached to the rotor shafts, so that the stored energy may be dissipated for test termination convenience and safety.

System accessories include three oil and vacuum subassemblies and associated line components. One subassembly is an oil heat exchanger with two air blower fan motors mounted thereon. A second package consists of a baseplate whereon an electric motor drives three lubrication oil pumps which serve the module flywheel bearings, clutches and shaft splines with oil. The third subassembly includes an electric motor which drives two scavenge oil pumps and a vacuum pump. Other filter and valve accessories are provided for installation in the lines between the pumps and the module.

System controls consist of five rack panels. A gas turbine control panel (GFP) is supplemented by a system control and monitor panel, a limits/alarm panel, a signal conditioning panel and a motor starter relay panel.

1.2 PROGRAM DESCRIPTION - The RS-31 program was initiated on 1 July 1975, at Rocketdyne, under Contract DAAG53-75-C-0278 to the U.S. Army Mobility Equipment Research and Development Command (MERADCOM) of Fort Belvoir, Virginia. The program is administered by the Electrical Discharge Laser Program Office (D/589) with technical direction by the Laser Sciences and Engineering Department, D/521, of Rocketdyne. After completion of a four-month analytical study on 1 November 1975, detail design was initiated and completed through critical design review on 8 June 1976. Hardware procurement and fabrication was completed during December of 1977, followed by final assembly of the basic module and the two air brakes at Rocketdyne.

Following completion of the test program, the RS-31 system is scheduled to be inspected, packaged and delivered to Ft. Belvoir, Virginia for systems integration and operation by MERADCOM. The air brakes are for test load purposes only, and will not be delivered.

1.3 TEST FACILITY DESCRIPTION - The RS-31 system is scheduled to be tested at Los Angeles Aircraft Division (Building #254) under the cognizance of the D/185 Thermodynamics Laboratory Test Function. The module, consisting of gas turbine, gearbox, rotor housing and brakes, shall be mounted from its three point base to the floor within test cell 109 of Building #256. The three oil subassemblies shall be placed nearby and plumbed to the module with placement and line orientation generally in accordance with the customer's intended installation arrangement, Fig. 7.2.

The five control panels shall be located in the adjacent control room, preferably with operator visual access to the module. Auxilliary systems and services provided by the facility shall be as described by Figures 7.3 and 7.4.

1.4 TEST PLAN - Testing of the system is described by Figure 7.5 where Tests A thru F consist of several sequences identified as 10,20,30... wherein speed is varied or held at plateaus as noted.

2.0 APPLICABLE DOCUMENTS

2.1 DRAWINGS

LE76-030-ER	RS-31 Assembly Drawing
LE76-031-ER	RS-31 System Test Interconnect Drawing
LE76-067-ER	RS-31 Circuit Diagrams
LE76-071-ER	RS-31 Mechanical Diagrams
LE76-224-ER	Air Turbine Brake Assembly, Left Side
LE76-225-ER	Air Turbine Brake Assembly, Right Side
2-000-001-34	T55-L-7C Engine Installation

2.2 SPECIFICATIONS

MIL-I-45208A	Inspection System Requirements
MIL-I-23699B	Lube Oil, Synthetic Base
124.31	Lycoming Engine Model Specification

3.0 REQUIREMENTS

3.1 CONTRACTUAL REQUIREMENTS - The system testing shall include spin-up to 100% of rated speed (15,000 rpm) with examination thereafter for defects. Instrumentation shall be inspected and calibrated in accordance with provisions of MIL-I-45208A.

3.2 FACILITY ASSEMBLY REQUIREMENTS

3.2.1 CONTROL PANEL ASSEMBLY

3.2.1.1 One motor starter panel assembled and wired per drawing LE76-066-ER shall be checked for continuity and functional operation of the starter relays using a signal to each starter relay coil as indicated.

3.2.1.2 One limit/alarm panel assembled and wired per drawing LE76-068-ER shall be checked for continuity and functional operation of the limit/alarm relay circuit using simulated sensor signal levels as indicated by Drawing LE76-067-ER.

3.2.1.3 One signal conditioning panel assembled and wired per Drawing LE76-069-ER shall be checked for continuity and correlation of the output signals to simulated applied input signals (as indicated by Drawing LE76-067-ER).

3.2.1.4 One control and monitor panel assembled and wired per Drawing LE76-070-ER shall be checked for continuity and functional performance of switches, relays, and lights, and the display meters shall be calibrated at 0 and 1.999 volts.

3.2.1.5 The four panels of para. 3.2.1.1 through 3.2.1.4 shall be mounted in a rack (along with one GFP gas turbine driver control panel) and interconnected in the order indicated by Drawing LE76-031-ER and LE76-067-ER.

3.2.2 Oil/Vaccum Subsystem Assembly - During assembly of the lube oil systems careful attention shall be directed to cleanliness of all parts and tubing to avoid contamination of the module oil jets.

3.2.2.1 One lube oil pumping subassembly assembled per Drawing LE76-032-ER shall be inspected for alignment and freedom of rotation.

3.2.2.2 One lube oil cooler subassembly assembled per Drawing LE76-033-ER shall be inspected for fan/motor freedom of rotation.

3.2.2.3 One lube oil scavenge and vacuum pumping subassembly assembled per Drawing LE76-034-ER shall be inspected for alignment and freedom of rotation.

3.2.3 FACILITY SUBSYSTEMS PREPARATION

3.2.3.1 A facility oil supply system shall be provided and interconnected to the RS-31 system, as defined by Drawings LE76-031-ER and LE76-071-ER. MIL-L-23699B oil shall be delivered with a 10 micron filter in the delivery line. The facility lube oil system is intended to serve the following functions:

a. To lubricate the air turbine brake bearings by providing 3.1 to 7.8 GPM to each brake at 60 to 400 psig. Since a small but undetermined amount of the oil will vaporize and leak into the turbine exhaust of the RH brake and be lost, a means should be provided to monitor oil sump level.

b. Toscavenge the air turbine brakes to remove and recirculate used oil.

c. To provide standby capability to deliver 7.5 GPM at 400 psig and/or 2.0 GPM at 50 psig in the event that supply pressure at the module input shall fall below those levels due to malfunction of the modules own supply system.

The AVCO gas turbine engine shall be lubricated by its on-board pumping system. MIL-L-23699B oil only shall be supplied for the engine reservoir to avoid any alternative oil in the test cell operation which could be mistakenly used for any other part of the system.

3.2.3.2 A facility vacuum system shall be provided and interconnected to the RS-31 system as defined by Drawings LE76-031-ER and LE76-071-ER. The facility vacuum system is intended to serve the following functions:

a. Evacuate the air turbine brakes to 2 $\pm$ 0.5 psia as limited by use of a controlled bleed valve.and line orifices.

b. Provide standby capability to limit pressure rise in the module's rotor case in the event of system pump malfunction or excessive case leakage beyond system pumping capacity.

3.2.3.3 A facility air system shall be provided and interconnect per drawings LE76-031-ER and LE76-071-ER. Command for application of air flow to the brakes shall be from module control panel LE76-070-ER and an air supply regulator shall be included to allow air pressure adjustment to about 135 psia at rated flow of 7 pps per brake. Inlet air flow on-off valves shall be arranged so as to avoid transient flow and pressure surges at start. A closed loop facility control system for brake air and vacuum systems shall be provided per LE76-067-ER, sheet 5

3.2.3.4 A facility JP-4 supply system shall be provided to deliver fuel to the gas turbine driver as indicated by Engine Spec. 124.31, Table 1.

3.2.3.5 With all module hardware in place, facility tubing and pipes shall be installed per Drawing LE76-031-ER. Precautions shall be taken to protect lines from damage and to secure fittings so that leakage and/or loss of flow could not results from vibrational and pressure forces.

3.2.3.6 Facility power shall be provided as listed below:

a) 208V, 3Ø, 400 Hz, 70 amps	Flywheel Module
b) 28V DC, 2.5 amps	Engine Control
c) 24V DC, 100 A-H battery	Engine Starter

3.2.4 INSTRUMENTATION SUBSYSTEMS

3.2.4.1 Sensors are generally supplied as a part of the RS-31 system except as identified in Table 8.1. Facility sensors shall also be provided as may be required for monitor and control of supporting facility subsystems such as standby-ready oil, vacuum and air as well as JP-4.

3-2-4-2 Some of the required display parameters are supplied with gages at the LE76-070-ER monitor/control panel, however, a brush and multipoint display/record capability is required and one facility digital speed meter is specified as indicated by Table 8.2.

3.2.4.3. Eight channels of brush and about 24 channels of tape are required as indicated by Table 8.2.

3.2.4.4 All required alarms and alarm lights are provided as a part of the RS-31 control package.

3.2.4.5 RS-31 systems cables are supplied to deliver vibration signals (M31 through M33) and vacuum signals (M20 and M21). All other instrumentation and control cables shall be provided by the facility and installed per LE76-031-ER.

3.3 MODULE INSTALLATION

3.3.1 The floor within test cell 109 shall be prepared so as to provide a base attached surface for the module which may be installed with or without baseplate LE76-232-ER (T4) as the interface. The module shall be oriented generally in accordance with Fig. 7.2.

3.3.2 Forward end closure materials shall be removed and the two brake assemblies shall be temporarily installed per drawings LE76-224-ER and LE76-225-ER. Verify that baffle plates LE76-058-ER are properly in place before the brakes are installed. Verify that brake rotational orientation is as specified by and that interface piping as required by LE76-071-ER will fit in the space available.

3.3.3 Aft end closure materials shall be removed and the Lycoming gas turbine shall be temporarily installed per drawing LE76-030-ER except shaft RES-1283 shall be omitted.

3.3.4 Remove plastic closure plug at LO-3 and install level switch EEPLS-2050 per LE76-071-ER.

3.3.5 Remove plastic closure plug at LO-26 and install oil heater MT6102 per LE76-071-ER.

3.3.6 Remove closure plugs and caps at ports LO-1,-2, -4, -7, -8, -9, -10, -11, -12, -13, -14, -16, -19 and install lube oil piping, valves, etc., per LE76-071-ER, Sheet 1.

3.3.7 Remove closure plugs at VAC-5A and VAC6A and install vacuum piping and valves, etc., per LE76-071-ER, Sheet 1.

3.3.8 Remove closure plugs at LO-20 through LO-25 and install facility lube oil system to the brake and to the coded module interfaces (22 and 23) per LE76-071-ER, Sheet 2.

3.3.9 Remove the closure plates at the brake air inlet and exhaust ports and install the facility air, vacuum and exhaust system per LE76-071-ER, Sheet 2. Also, connect the facility vacuum system to coded module interfaces (20 and 21).

3.3.10 Install all system sensors per Table II and cable to the display system per LE76-031-ER and LE76-067 through 070-ER.

3.3.11 Install all system control cables per Table II and III and LE76-031-ER and LE76-066 through 070-ER.

3.4 TEST REQUIREMENTS

3.4.1 POWER CIRCUITS TESTS (Ref. Drawing No. LE76-066 through 070-ER.)

3.4.1.1 With all circuits complete except AC source power, verify open circuit between: a) T1-30 and F1-A c) T1-30 and F1-B
b) T1-30 and T1-27 d) T1-30 and F1-C

when S1 is open.

3.4.1.2 Close S1, Verify continuity between:

a) T1-30 to T1-27, T4-1 and M18-1
b) T1-28 to T1-3, T1-8, T4-3 and M18-3
c) T1-29 to T4-2 and M18-2

Check to verify that (b) and (c) are also true with S1 open.

3.4.1.3 With S1 closed verify open circuit between T1-30 and all other terminals of T1 and T2.

3.4.1.4 Verify open circuit between M1-1 and all non-1 terminals of LE76-068-ER.

3.4.1.5 With all switches in the off position, apply AC power to T3, turn S1 on, and verify the following:

a) Alarm lights L3C, L7C, L8C, L6B and L7B are on.
b) T2-32, T2-40, T2-45, and T2-47 register + 5VDC.
c) Displays D1 through D4 and D9 through D12 are on.
d) T1-1 and T1-2 read 120VAC.
e) T1-4, T1-5, T1-6 and T1-9 through T1-24 are 0 VAC.
f) M29 through M33 deliver $\pm$ 15VDC.

3.4.1.6 Jumper from T2-33 to R4-3 and cerify alarm lights L6B and L7B go off.

3.4.1.7 Close S7, reset S6 and cerify trip light L7B goes off.

3.4.2 COMMAND CIRCUIT TESTS Ref. dwgs LE76-066 thru 070-ER

3.4.2.1 Turn switch S1 on and cerify actuation of:

a) Vacuum valves from S10, S11
b) Brake valves from S4
c) Oil cooler fans from S3
d) Trip from remote switch (open jumper T1-16 from T1-17)

3.4.2.2 Verify conditions as follows:

a) T1-23 to T1-24 is shorted.
b) T1-21 to T1-22 is shorted.
c) T1-18 to T1-19 is open and T1-19 to T1-20 is shorted when trip light (L7B) is off and reversed when light is on.
d) T1-13 to T1-14 is open and T1-14 to T1-15 is shorted when alarm light (L6B) is off and reversed when light is on.
e) T2-34 to T2-35 is shorted when reset switch S6 is depressed.

3.4.2.3 Momentarily actuate switch S2 to verify that lube and vacuum pump motors are activated but do not allow motors to continue operation unless lube oil has been provided to inlet and provisions have been made to systematically check out lube system per para. 3.3.4.1.

3.4.3 SIGNAL CIRCUIT TESTS Ref dwgs LE76-066 thry 070-ER

3.4.3.1 Apply pressure to oil pressure switches M35, M36, and M37 to verify actuation at the specified set pressure as pressure decays.

3.4.3.2 Short chip detector M38 to verify operation of alarm light L6B.

3.4.3.3 With all systems activated make the following tests to check limit alarm functions:

a) Apply frequency signal to circuit M1 and M2 to check operation of M3, M4, M5, M6, R9 and R10. Verify reading at D1 and D12 displays.

3.4.4 SUBSYSTEM TESTS

3.4.4.1 Lubrication, Air and Vacuum Subsystems. Ref. dwg LE76-071-ER

3.4.4.1.1 Verify completion of installation including facility flowmeters in lines 10, 11, and 12.

3.4.4.1.2 Remove breather at L017 and fill sump through L017/L018 with 8.35 gallons of clean MIL-L-23699B lube oil.

3.4.4.1.3 Remove plug L015 and add two gallons of oil directed so as to flow into the deaerator sump and L013.

3.4.4.1.4 Loosen the strainer cap at the line 4 strainer and bleed air as required to allow oil up to pump E, then retighten cap.

3.4.4.1.5 Loosen the line plug at pump A to bleed air until oil flows by gravity to the A, B and C pump interfaces and retighten plug.

3.4.4.1.6 Verify that the vacuum pump is filled with vacuum pump oil to middle of oil sight window.

3.4.4.1.7 Using control panel switches S1 through S3, activate the module oil/vaccum (o/V) subsystems and verify oil system parameters. Visually observe aft case flows through port L017. Check for leakage at joints. Verify proper TRV and PRV control. Record evacuation rate at D7 and D8.

3.4.4.1.8 Turn off module O/V system and check filters and vacuum trap for signs of contamination and rotor seal leakage.

3.4.4.1.9 Continue para. 3.4.4.1.7 as necessary to establish cleanliness of system and evacuation pressure stabilization. Check and clean filters as required.

3.4.4.1.10 Checkout facility standby-ready oil system during operation by cutting off lube oil pump switch S2 and scavenge/Vacuum switch S3, and vacuum valve switches S10, and S11, to close off valves. Visually monitor oil rise through L018 into clutch case and record vacuum decay rate. After 10 minutes turn facility oil off and S3 on to scavenge oil and check vacuum pump limit vacuum with intake valves closed.

3.4.4.1.11 Checkout facility oil system flow to air turbine brakes to flush system and to set flow and pressures.

3.4.4.1.12 Checkout facility vacuum system to draw vacuum on air turbine brakes with brake 1 be oil flow on and record final pressure achieved. With vacuum and oil flow on to brakes setup air system to standby-ready state and verify ready-to-brake condition.

3.5 OPERATIONAL TESTING

3.5.1 TEST OBJECTIVES

3.5.1.1 Primary - The primary objective of operational testing shall be to demonstrate the capability of the RS-31 system to store and deliver energy at speeds between 10,500 and 15,000 rpm..

3.5.1.2 Secondary - The secondary objectives of operational testing shall be to:

a) Characterize RS-31 system normal operating parameters such as:

1. Rotor gas versus speed
2. Case vacuum pressure range
3. Rotor vibration versus speed
4. Oil heating relationships
5. Parasitic losses
6. First critical speed and vibration

b) Characterize normal RS-31 system functional relationships:

1. Oil system capability in terms of : filtration, de-aeration, scavenging, sump capacity, priming, temperature control

2. Vacuum and seal capability in terms of: shaft leakage, pump down rate, ultimate vacuum, windage heating

3.5.2 OPERATIONAL TESTS

3.5.2.1 TEST A - Test A is for operation checkout and operation experience with the engine only. The nine sequence steps of test A shall be performed until it is clear that operational control is without difficulty of any kind.

3.5.2.2 TEST B - Test B serves to checkout the air brake control system and to provide preliminary verification of flywheel rotational integrity including clutches in the non-engaging mode. Test B may be repeated several times until satisfactory operation is verified and may include direct in-call observation at the specified speeds as well as possible higher speed rotation up to but not into the first critical speed regime.

3.5.2.3 TEST C - Test C is to be the first of the complete system operational runs using all subsystems. The primary goal is to characterize operation thru the first critical speed. Following Test C, a data review and hardware inspection will be required before proceeding to test D with emphasis on defining any desireable modifications to the performance of auxilliary hardware instrumentation, and/or operating procedures.

3.5.2.4 TEST D - This test will repeat Test C with more cycles and longer dwell and shall serve to provide an initial evaluation of clutch re-engagement for prediction of higher speed performance. Test D may be repeated at 70° F oil temperature to predict bearing friction losses at 14000 rpm and at 100 Torr case pressure to simulate windage losses at 14000 rpm.

3.5.2.5 TEST E - Test E brings rotor speed to the lower bound of the rated operating range, verifies clutch operation at this point and serves to establish calibration points for the strain detection system where rotor disc radial growth is at 0.025 inches . Accurate assessment of rotor and ring temperature versus time and speed will be sought to assist the calibration and set reference values. Test E may be performed at alternate oil temperature and case pressure to verify parasites loss and energy extraction assumptions. If time permits, test E operation may include an elevated pressure windage deceleration for evaluation of heat rise and decay rate. Brake usage at sequence 20/30 is provided for confidence in its performance for emergency use. At sequence 50 the brake will be used in conjunction with engine drive torque as a calibration device for assessing power brake and module energy during deceleration.

3.5.2.5 TEST F - This test evaluates system performace to full speed with a brief expansion to the limit speed of 15,600 rpm. Recharge cycles are schedule to verify proper system performance under conditions similar to normal field usage in the 10,500 to 15,000 rpm range. Use of the brake at sequence 100/110 is planned to calibrate rotor inertia stored energy relationships.

3.5.3 Operational Procedure Ref. Figure 7.6

3.5.3.2 Activation

3.5.3.2.1 Module Controls

a) Verify that switches S1 thru S11 are all OFF ____________

b) Turn power switch S1 ON ____________

c) Verify that alarm lites L3C, L4C, L5C, L7C, L8C and L6B are all ON ____________

d) Press - test lites L5B, L7B, L1C, L2C, & L6C ____________

e) Press reset switch (S6) and verify that alarm lite L6B goes off ____________

f) Turn lube/vac switch (S2) ON ____________

g) Turn vacuum valve switches S10 & S11) ON ____________

h) Turn oil cooler motor swtich (S3) ON ____________

i) Command facility oil, air and vacuum systems ON ____________

j) Monitor lites until L3C, L4C, L5C, L7C and L8C go OUT ____________

3.5.3.2.2 Engine Controls

a) Verify that all swtiches are in the OFF or DECREASE position ________

b) Clear the test area of personnel ________

c) Turn power switch ON ________

d) Turn the fuel switch ON ________

e) Turn the igniter switch ON ________

f) Turn the starter switch ON ________

g) When the engine exhaust temp. reaches 700^{o}C turn the igniter switch OFF ________

h) When N_1 speed reaches 40% turn the starter switch OFF ________

i) Use switch N_1 to adjust torque or rate of module speed increase and N_2 to limit speed to target values during test. ________

<u>WARNING</u> - Where AVCO engine is left on to counter speed decay the operator shall constantly remain in active control of speed.

3.5.3.3 <u>NORMAL SHUTDOWN</u>

1. Turn engine fuel Switch OFF ________
2. Turn brake switch (S4) ON if specified ________
3. When module speed has decayed to Zero:

   a) Turn brake switch (S4) OFF ________

   b) Turn facility systems OFF ________

   c) Turn oil pump switch (S2) OFF ________

   d) Turn vacuum valve "open" switches (S9 & S10) OFF ________

   e) Turn scavenge/vacuum pump switch (S3) OFF ________

NOTE: Do not turn controller power switch OFF unless loss of case vacuum is acceptable.

3.6 EMERGENCY CRITERIA

3.6.1 Emergency Shutdown (ESD)

3.6.1.1 An emergency shutdown is accomplished exactly like normal shutdown (NSD) per paragraph 3.5.3.3, except that Step 2, brake switch (S4) activation is mandatory. ESD verbal command shall be "Cut and Brake."

3.6.1.2 If ESD is initiated due to loss of module vacuum the NSD shutdown procedure shall be augmented as follows:

Step 2.1 Open facility backup evacuation valves

Step 2.2 Close module evacuation valves by turning switches S9 and S10 off. The ESD command is this case shall be: "Cut, Brake and Go to facility vacuum.

3.6.1.3 If shutdown is only related to brake problems and braking could be detrimental normal shutdown (NSD) shall be employed.

3.6.2 Drive shutdown (DSD)

Where problems are encountered solely in the engine, gearbox and in cases of brake malfunction, drive shutdown only may be suitable without immediate forcing module speed down by braking. Command "Engine Fuel OFF" for drive shutdown only.

3.6.4 Parametric Decisions

A total of 34 parameters will be displayed during system operation as indicated by Table 8.2. A test parameter monitor shall be assigned to each of the four display stations (in addition to the test operator). The test conductor and project representative shall be free of operational duties so as to be able to take note of any detected deviation, however, the test monitors shall independently command shutdown wherever and immediately after any parametric limit is exceeded. Ref. Note 6.2.1

3.6.5 Redline Values

3.6.5.1 Brush Recorder (Station 1)

The 8 parameters on Brush as noted in Table 8.2 are all related to lubrication of bearings and gears. The 4 module rotor bearing temperatures are actually sensed as oil temperature in the respective bearing cavity exit flow line and are predicted to correlate closely one to another, running about 10 to 50°F above oil temperature M10 displayed at gage D9. Change should be very slow. Any individual increase of more than 10°F indicates restricted flow or excessive friction and is cause for ESD. Three oil flow rates on Brush recorder will be constant in value (Brake M59, Forward M49, and Aft M50), and any deviation of more than ±10% (0.2 GPM, 0.25 GP, and 0.75 GPM, respectively), indicates flow restriction or leakage and is cause for ESD. Gearbox flow will be proportional to speed at 0.6 GPM per 1000 RPM. Any deviation of more than 10% from nominal value at operational speed is cause for DSD (only) with further monitor of gearbox speed required to assure that DSD results in declutch and gearbox speed deceleration to zero RPM.

3.6.5.2 Multipoint Recorder (Station 2)

The 12 parameters displayed on multipoint are all temperatures of bearing outer races and should correlate closely to each other and increase with speed and oil temperature M10 at gage D9. Deviation of any one or more race temperature from temperature of the same set by more than 10°F shall be cause for ESD. Three alternate temperature parameters may be recorded on multipoint during the test series as noted on Table 8.2 with redline values to be pre-defined at the time of use.

3.6.5.3 Operator Displays (Station 3)

3.6.5.3.1 Module Control Panel

A set of 10 meters at the module operator's panel indicate rotor speed (2), gap$^{(2)}$, vibration $^{(2)}$, as well as cavity temperature $^{(1)}$ and pressure $^{(2)}$ and oil supply temperature $^{(1)}$. Redline values for these parameters will generally vary from test to test as a function of test goals. Predicted redline criteria for each test are listed in Table 8.3. Note that 14 measurements are handled by these 10 meters and all are preset to activate the panel alarm buss and alarm lite L6B as discussed under paragraph 3.6.4.3.3.

3.6.5.3.2 Engine Control Panel

A set of 4 meters at the engine control panel (Figure 7.6.2) display engine torque, exhaust temperature and speed . Torque shall never be allowed to exceed 1000 ft. lbs. Engine exhaust temperature should never exceed 800°C and speed levels should not exceed the D1/D12 values of Table 8.3. In the case of N1, (the first state rotor) each 10% corresponds to 1500 RPM.

3.6.5.3.3 Limit Alarms

3.6.5.3.3.1 Alarm Display, L6B

A set of 19 parameters are linked to alarm light L6B such that any of the 19 will activate L6B if the preset alarm limit value is exceeded. Table VI 8.4 describes the individual set points for alarm modules and switches for each test. Category B through G alarms latch and may be identified subsequently by alarm lights on each source module. Category A, H and I alarms do not latch but are identified by individual panel lights L3C through L5C, L7C and L8C.

Oil pressure alarm lights activate in a first out mode but are series regressive. Common alarm light L6B does not latch but after activation by a latching module, the L6B alarm condition may be released by reset switch, S6, after correction.

Activation of a preset alarm shall not be a required ESD condition but shall serve as a warning of a momentary deviation or a continuing problem requiring verification by other means.

3.6.5.3.3.2 Alarm Display, L6C

Module oil sump outlet quality is monitored by a magnetic particle (chip) detector. Activation of L6C shall be followed by judgment based on other oil pressure and flow parameters as to need for ESD. In any event, lube system service

is required prior to further test.

3.6.5.3.3.3 Alarm Display, L1C

Module oil sump level is monitored by a horizontal acting lever switch which activates console light L1C at minimum allowable level. Activation of L1C shall result in ESD.

3.6.5.4 Facility Displays (Station 4)

A set of 6 bourdon gages are provided to display air and oil pressures as noted by P1 through P6 of Table 8.2. Nominal and redline values are as follows:

	NOMINAL	REDLINES
P1	2 psia	<1/>5 psia
P2	120 psia	SEE BELOW
P3	500 psig	<425 psig
P4	60 psig	< 40 psig
P5	60 psig	< 40 psig
P6	0-40 psig	10% LOW

In the case of brake air displayed on gage P2, the reading should be 15 psia, except upon command for brake operation and 120 psig ± 10 at that time. Under ESD conditions a P2 deviation is not of primary significance, however, where brake use is not critical to safe shutdown excessive brake pressure shall be resolved by deletion of brake air flow. (Open switch 54). In the case of variable gearbox oil pressure at P6, a curve of speed vs pressure (Fig. 7.7) shall be located near the gage for redline action and values less than indicated shall be reacted by DSD.

One digital speed meter, P7, at the facility console, shall display gearbox speed. Redline values for gearbox speed shall be per Table 8.3 as specified for module rotor speed. The primary application of the P7 display shall be for monitoring acceleration and deceleration of the drive when not engaged to the module rotors.

3.7 INSTRUMENTATION

Table 8.5 describes the complete instrumentation system.

4.0 QUALITY ASSURANCE

Instrumentation shall be inspected and calibrated in accordance with the requirements of MIL-I-45208A.

5.0 DISPOSITION OF HARDWARE

6.0 NOTES AND DATA

6.1 DATA

6.1.1 Properties of MIL-L-23699B Oil

6.1.1.1 Specific Heat, BTU/lb - ^{o}F

6.1.1.2 Thermal Conductivity, BTU-ft/ft^2 - HR - ^{o}F

@ $200^{o}F$ .0859

6.1.1.3 Viscosity, Centistakes and SSU

@ $50^{o}F$	150 C_s	680 SSU
70	70	324
100	28	132
150	10	57
200	4.6	41

6.1.1.4 Density, lbs/ft^3 and lbs/gal

@ $60^{o}F$	63.18 lbs/ft^3	8.45 lbs/gal
100	61.58	8.23
150	60.15	8.04
200	58.71	7.85

6.1.1.5 Dielectric Constant

@ $79^{o}F$ 3.73 (at 1000 Hz)

6.2 NOTES

6.2.1 Designated Operator - A qualified controls operator(s) shall be designated by the manager of test (D.185) and shall be the sole person(s) authorized to operate the RS-31 controls and in particular the driver engine (AVCO gas turbine) which controls speed.

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Pg 18

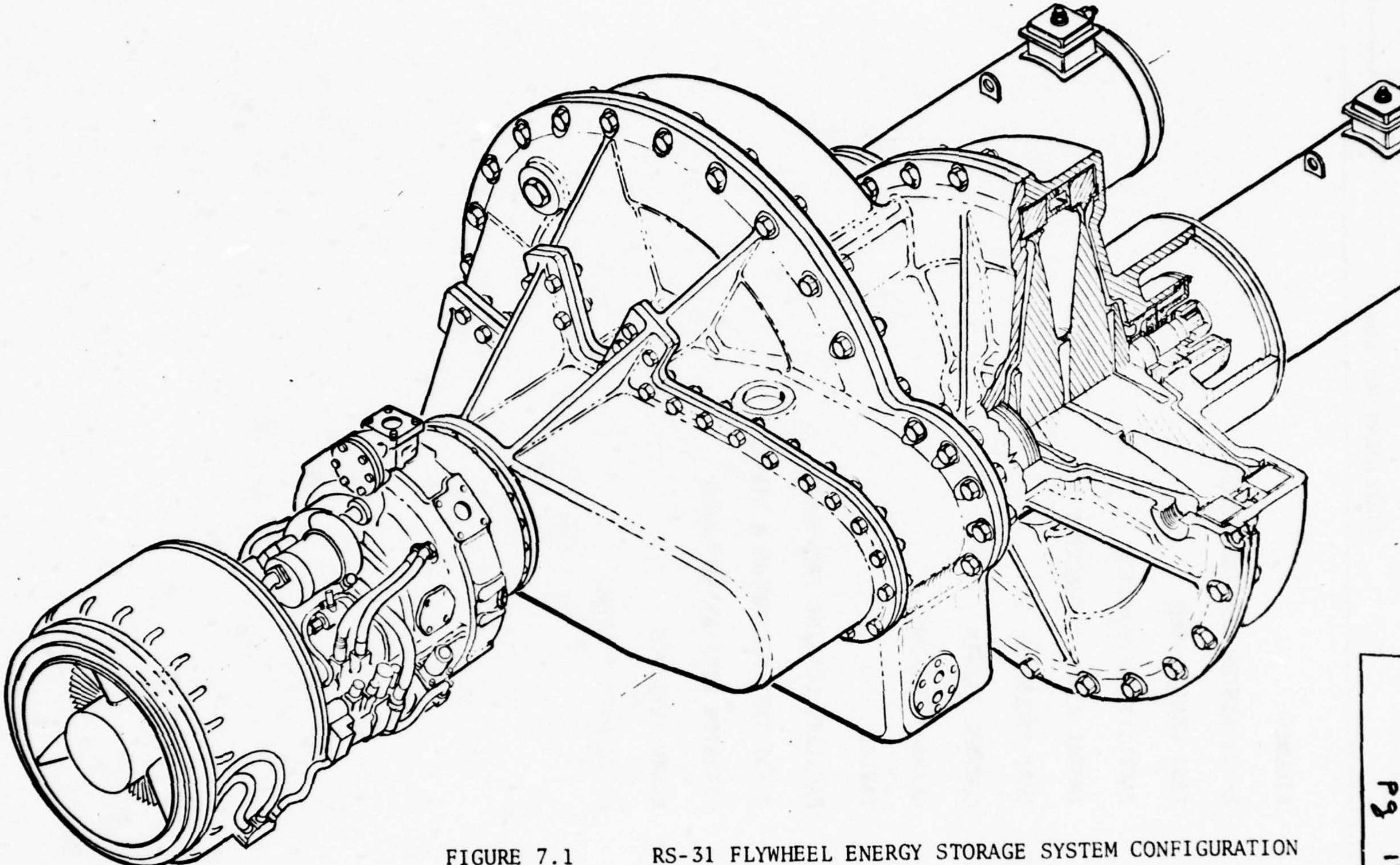

FIGURE 7.1 RS-31 FLYWHEEL ENERGY STORAGE SYSTEM CONFIGURATION

Rockwell International Corporation
Rocketdyne Division
Canoga Park, California

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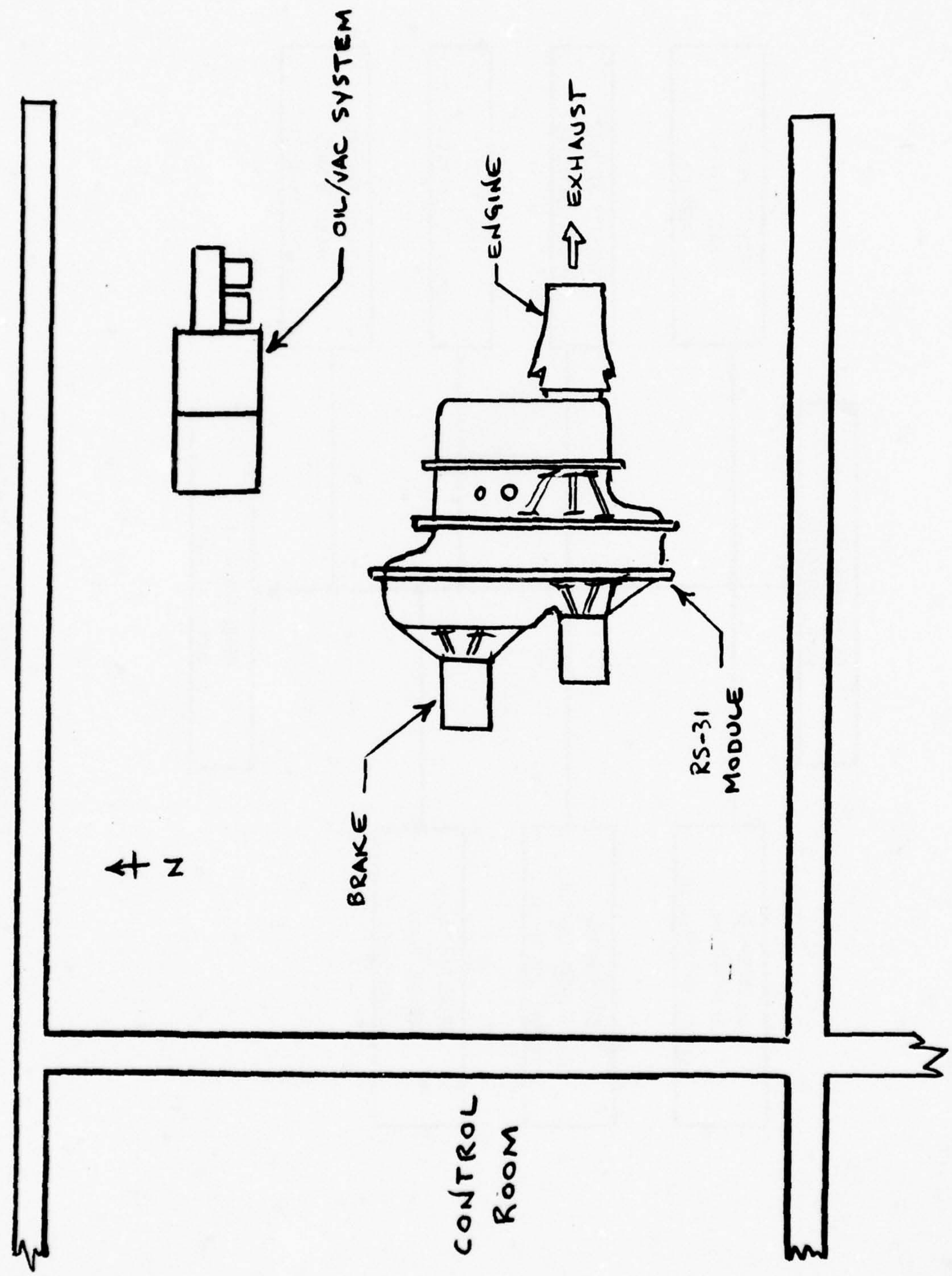


Figure 7.2 Cell Layout

RC-1290A

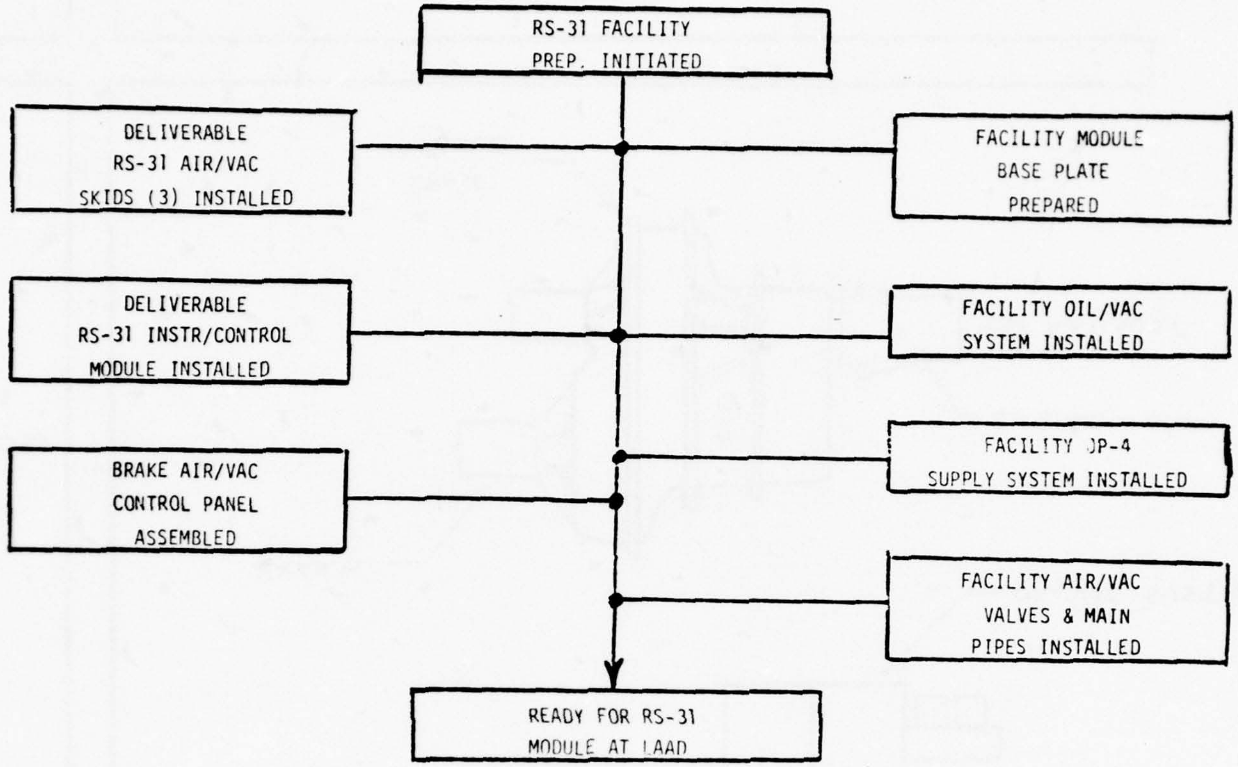


Figure 7.3. Facility Preparation

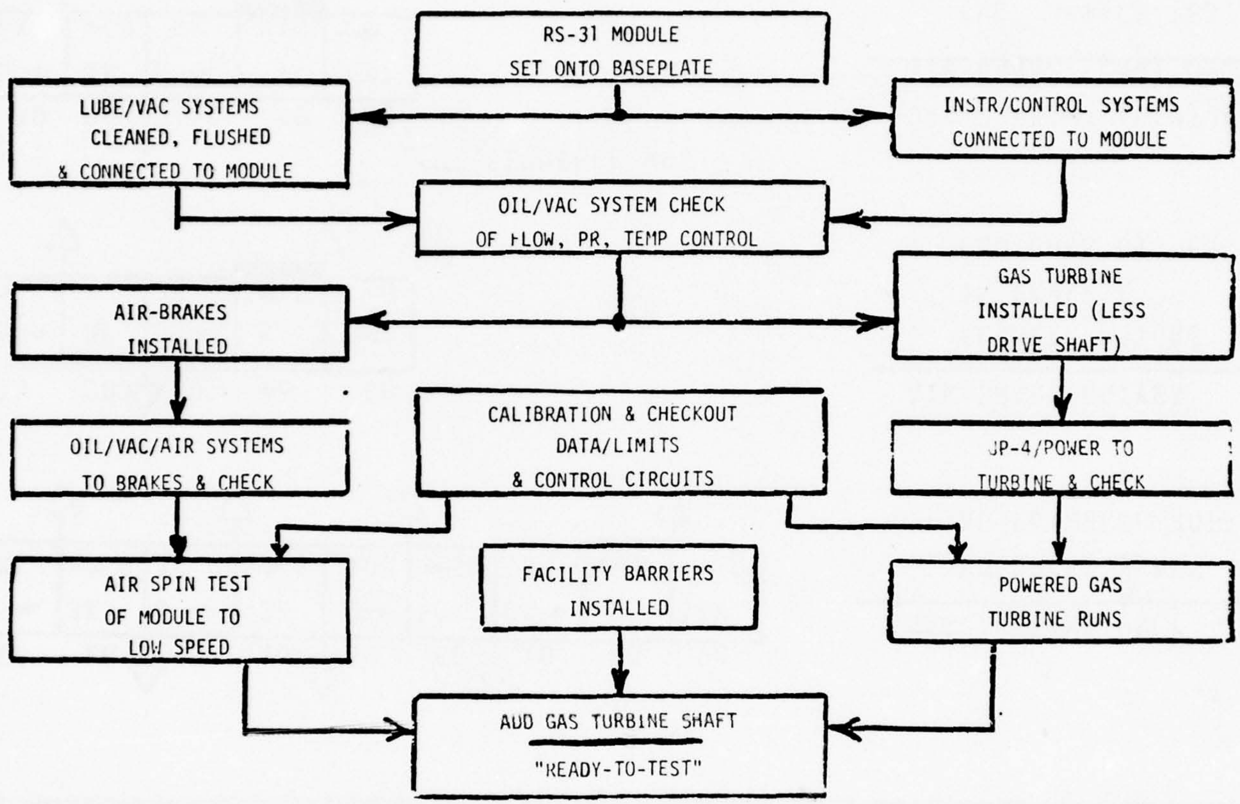


Figure 7.4. Installation & Checkout

RC-1290A

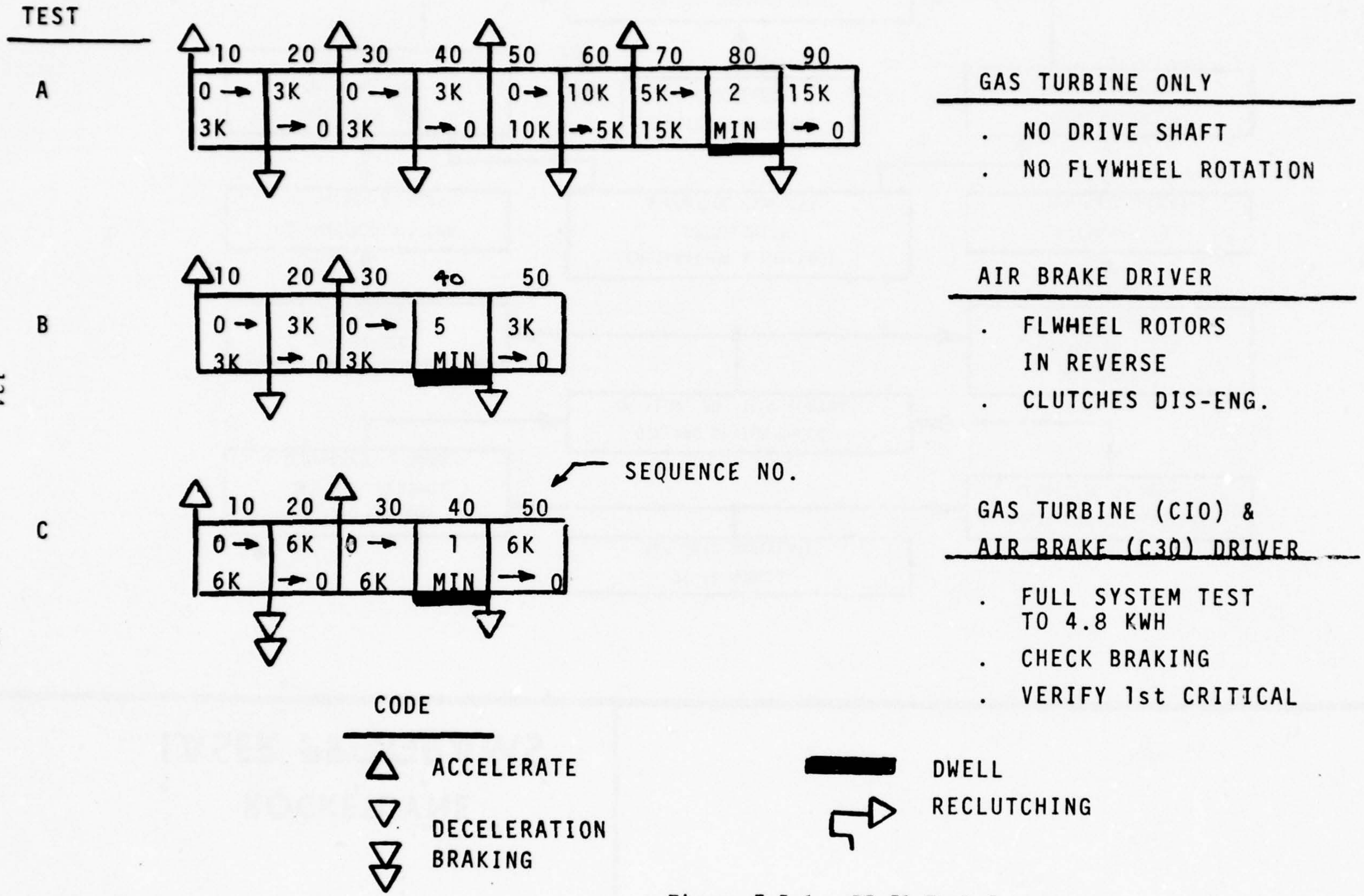


Figure 7.5.1. RS-31 Test Sequence

RC-1290A

GAS TURBINE DRIVE

- STEADY STATE DATA
- DEMONSTRATE RECLUTCHING ACCELERATION
- CHECK 15 KWH OPERATION
- MEASURE LOSSES
- DEMONSTRATE RATED SPEED & DESIGN MARGIN TO 32.5 KWH

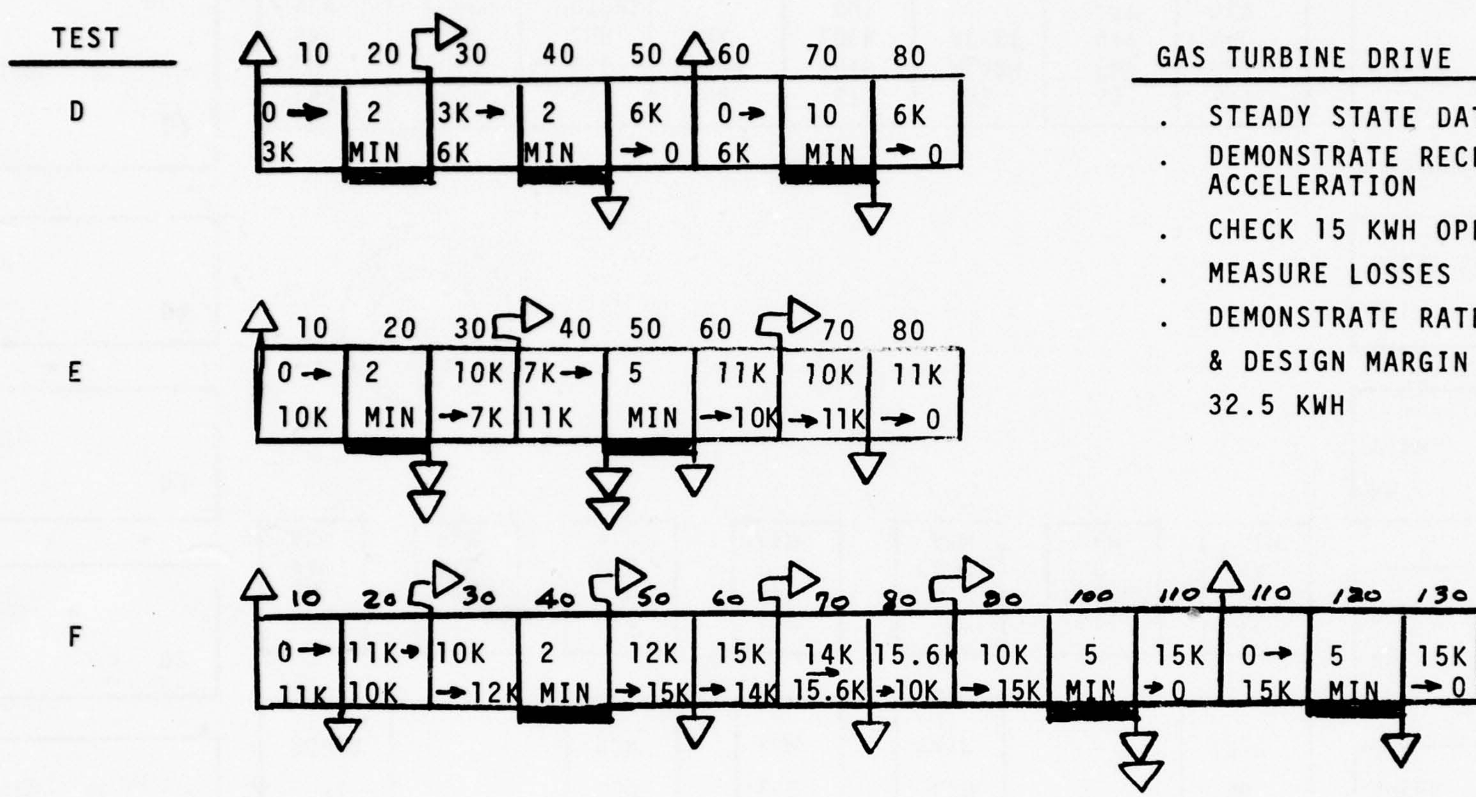


Figure 7.5.2. Test Sequence (Continued)

RC-1290 A

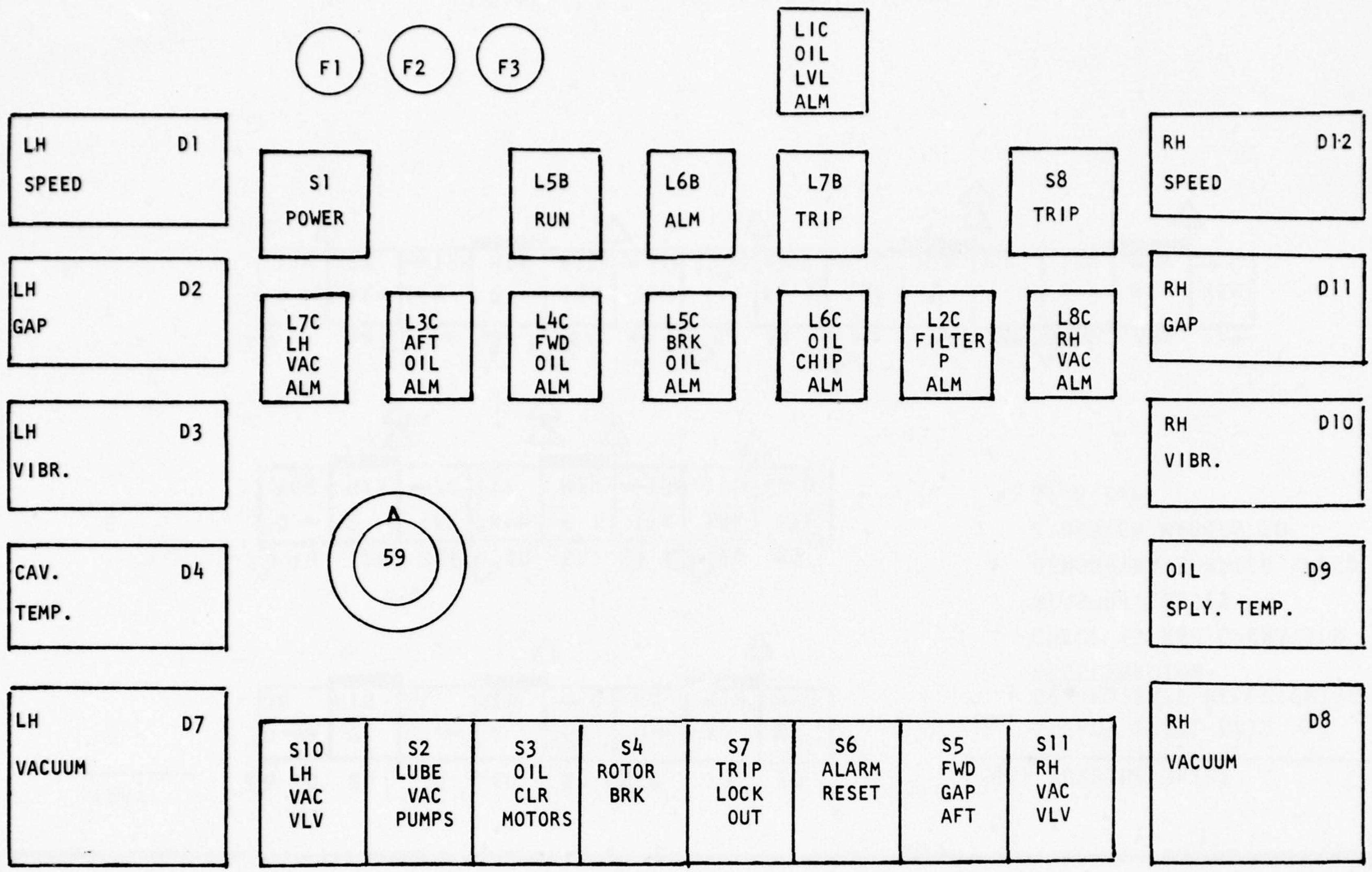


Figure 7.6.1. OPERATORS CONTROL PANEL

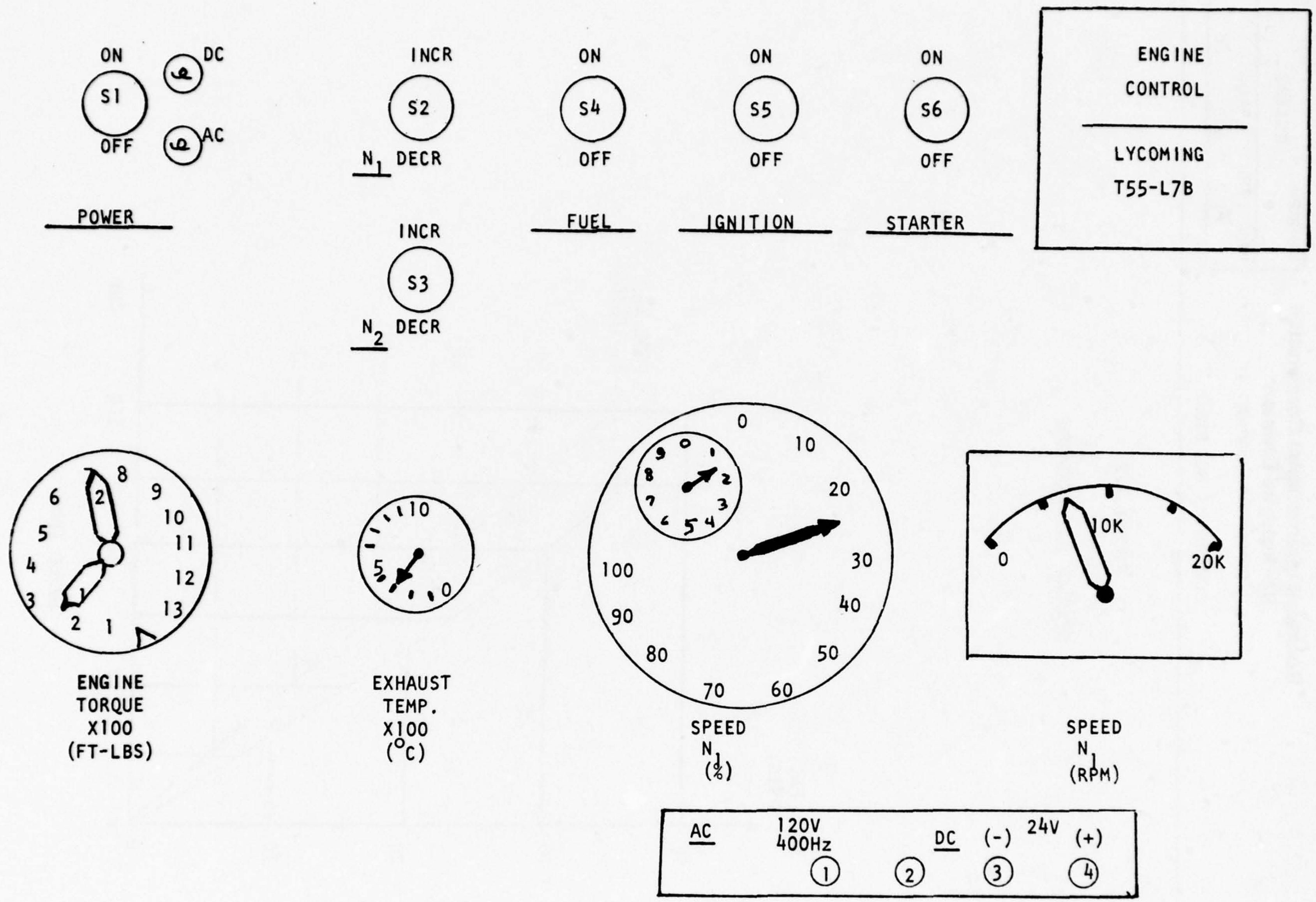


Figure 7.6.2. ENGIRE CONTROL PANEL

FIGURE 7.7

GEARBOX OIL PRESSURE

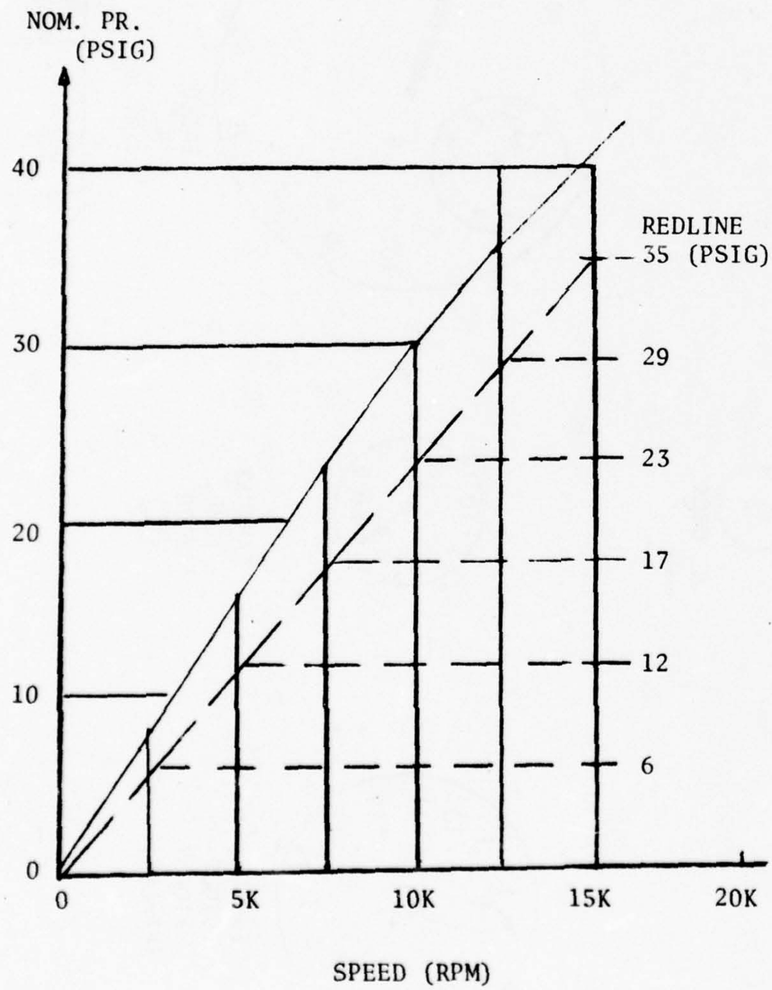


TABLE 8.1

FACILITY SENSING PROVISIONS

m1 → m21	NONE
m22 → m24	(a) 3 pressure sensors to recording system
	(b) 3 pressure lines to panel gages
m25 → m44	NONE
m45 → m48	(a) 4 Endevco 2220 C accels. to recording system
m49 → m50	(a) 2 flow sensors to recording system
m51 → m54	(a) 4 pressure sensors to recording system
m55	(a) 1 DC vacuum pressure sensor to control circuit and recording system
m56	(a) 1 DC pressure sensor to control circuit and recording system - (PX2BC sensor will serve objectives)
m57	(a) 1 pressure line to gage panel
m58 → m69	NONE
m70 → m71	(a) 2 skin temps to recorder

TABLE 8.2

RS-31 TEST RECORDING & DISPLAY

STATION 1 (8)

BRUSH RECORDER

B1	RR BRG TEMP	m4
B2	LR BRG TEMP	m5
B3	RF BRG TEMP	m6
B4	LF BRG TEMP	m7
B5	BRAKE OIL Q	m59
B6	GB OIL Q	m60
B7	FWD OIL Q	m49
B8	AFT OIL Q	m50

STATION 2 (12)

MULTIPOINT RECORDER

MP1	BF BRG #1 F TEMP	m61
MP2	GB BRG #1 R TEMP	m62
MP3	GB BRG #2 F TEMP	m63
MP4	GB BRG #2 R TEMP	m64
MP5	GB BRG #3 F TEMP	m65
MP6	GB BRG #3 R TEMP	m66
MP7	GB BRG #4 F TEMP	m67
MP8	GB BRG #4 R TEMP	m68
MP9	LF BRAKE TEMP	m41
MP10	LR BRAKE TEMP	m42
MP11	RR BRAKE TEMP	m43
MP12	RF BRAKE TEMP	m44

ALTERNATIVES

MP13	GB OIL TEMP	m69
MP14	GB CASE TEMP	m70
MP15	GB CASE TEMP	m71

STATION 3 (22)

OPERATOR DISPLAYS

D1	LH SPEED	m2
D2	LEFT GAP	m27/29
D3	LEFT VIBR	m31/33
D4	LF CAV T	m9
D5		
D6		
D7	LF CAV PR	m21
D8	RF CAV PR	m20
D9	AFT OIL T	m10
D10	RIGHT VIBR	m30/32
D11	RIGHT GAP	m26/28
D12	RH SPEED	m1

ALARM LIGHTS

L1C	OIL LEVEL	m40
L2C	FILTER P	m39
L3C	AFT OIL PR	m35
L4C	FWD OIL PR	m36
L5C	BRAKE OIL PR	m37
L6C	CHIP DET	m38
L7C	LH CAV VAC	m21
L8C	RH CAV VAC	m20

ENGINE CONTROL DISPLAYS

D21	TORQUE	m16
D22	EXH TEMP	m17
D23	SPEED %	m18
D24	SPEED RPM	m19

STATION 4 (7)

FACILITY DISPLAYS

P1	BRAKE VAC PR	m55
P2	BRAKE AIR PR	m56
P3	AFT OIL PR	m23
P4	FWD OIL PR	m22
P5	BRAKE OIL PR	m57
P6	GB OIL PR	m24
P7	GEARBOX SPEED	m3

STATION 5 (28)

TAPE RECORDER
(NOT DISPLAYED)

T1	RH SPEED	m1
T2	LG SPEED	m2
T3	GB SPEED	m3
T4	RF CAV T	m8
T5	LF CAV T	m9
T6	CLR IN T	m11
T7	CLR OUT T	m12
T8	BRAKE AIR PR	m56
T9	BRAKE VAC PR	m55
T10	FWD OIL PR	m22
T11	AFT OIL PR	m23
T12	GB OIL PR	m24
T13	RF GAP	m26
T14	LF GAP	m27
T15	RR GAP	m28
T16	LR GAP	m29
T17	RF VIBR	m30
T18	LF VIBR	m31
T19	RR VIBR	m32
T20	LR VIBR	m33
T21	LF BRK VIBR	m45
T22	LR BRK VIBR	m46
T23	RF BRK VIBR	m47
T24	RR BRK VIBR	m48
T25	RR BRK PR	m51
T26	RF BRK PR	m52
T27	LR BRK PR	m53
T28	LF BRK PR	m54

TABLE 8.3

OPERATOR DISPLAY REDLINES

TEST	NOM SPEED	SPEED D1/D12	GAP D2/D11	VIBR D3/D10	VACUUM D7/D8	CAV. T. D4	OIL T. D9
	RPM	RPM	IN.	Gs	mm Hg	^oF	^oF
A	3000	3600	.010	1.5	-	150	150
B	3000	3600	.010	1.5	-	150	150
C	6000	7000	.010	3	20	150	150
D	6000	7000	.010	6	20	200	150
E	11,000	11,500	.030	6	15	200	150
F	15,000	16,000	.060	10	10	250	200
MEAS		M1	M26	M30	M20	M9	M10
		M2	M27	M31	M21		
			M28	M32			
			M29	M33			
ALARM		M1	M13	M17	D7	M11	M12
		M2	M14	M18	D8		
			M15				
			M16				

TABLE 8.4

ALARM BUSS/L5B LIGHT SETTINGS

TEST	VAC D7/D8	SPEED M3/M5	BRG. T M7-M10	CAV. T. M11	OIL T. M12	GAP M13-M16	VIBR M17/M18	OIL PR PS #1	OIL PR PS 2,3
	mm Hg	RPM	°F	°F	°F	IN.	G's	PSIG	PSIG
A	20 (1)	3600	200	150	150	.010	1.5	425	40
B	20 (1)	3600	200	150	150	.010	1.5		
C	20 (2)	7000	200	150	150	.010	3.		
D	20	7000	200	200	150	.010	6		
E	15	11,500	250	200	150	.030	6		
F	10	16,000	250	250	200	.060	10		
CATE-GORY	A	B	C	D	E	F	G	H	I

NOTES: (1) SET FOR CHECKOUT PURPOSES ONLY. ACTIVATES L7C and L8C ALARM LIGHTS. JUMPER BUSS TO ABOID L5B ALARM.

(2) REMOVE JUMPER AS NOTED ABOVE.

DATA SHEET TABLE 8.5 RS-31 DATA SYSTEMS

		RH SPEED	LH SPEED	GB SPEED	RR BRG TEMP	LR BRG TEMP	RF BRG TEMP	LF BRG TEMP	RF CAV. TEMP
MEAS. NO.		m1	m2	m3	m4	m5	m6	m7	m8
CODE		N-RH	N-LH	N-GB	T-RR	T-LR	T-RF	T-LF	T-RC
SENSOR	MFG	ELECTRO	→	→	RC1282	→	→	→	→
	P/N	603384	→	→	-3C	-8C	-3C	-3C	-3S
	RANGE	0-16K RPM	→	→	0-300F	→	→	→	→
	TAP THRD	5/8-18	→	→	1/8 NPT	→	→	→	→
	TAP DPTH	0.947	→	2.400	1.563	5.000	1" LINE TEE		1.200
	LOCATION	MIDDLE CASE	AFT CASE	GEAR BOX	MIDDLE CASE	AFT CASE	OIL LINE #2A	OIL LINE #2B	FWD CASE
	DRAWING	026	025	ROK 8	026	025	071	071	027
	ZONE	A47	F09	×	B47	E48	SHT 1	SHT 1	F21
	VIEW	HH	CC	FF	EE	FF	10-11	10-12	BB
CABLES	SENSOR	F 3/c SH	F →	F →	F Fe/Cn	F →	F →	F →	F →
	DISPLAY #	m1S	m2S	m3F	×	×	×	×	×
	-FROM	M1	M2	sensor	×	×	×	×	×
	CALIBR #	m1A m1H, m1L	m2A m2H, m2L	×	m4A	m5A	m6A	m7A	×
	-FROM	M3 M4, M4	M5 M6, M6	×	M7	M8	M9	M10	×
	RECORD #	m1F	m2F	m3F	m4F	m5F	m6F	m7F	m8F
	-FROM	M1	M2	P7	M7	M8	M9	M10	SENSOR
SWITCHING	CALIB.	9-12B -12C, -12D	9-1B -1C, -1D	×	9-4C	9-4D	9-9C	9-9D	×
	DISPLAY	9-12A	9-1A	×	×	×	×	×	×
OUTPUT	DISPLAY	D12	D1	P7	B1	B2	B3	B4	×
	-VOLTS	0-2 VDC	0-2 VDC	0-2667 HZ	mV	mV	mV	mV	×
	RECORD	T1	T2	T3	B1	B2	B3	B4	T4
	-VOLTS	0-2667 HZ	0-2667 HZ	0-2667 HZ	mV	mV	mV	mV	mV
	ALM. LITE	L3M L4M(2)	L5M L6M(2)	×	L7M	L8M	L9M	L10M	×
NOM. RTD. VALUE		10.5 → 15K RPM	→		200°F	→	→	→	212°F
OBS. RED LINE		D12	D1	P7	B1	B2	B3	B4	×
ALARM	LIMIT	M3-HI	M5-HI	×	M7-HI	M8-HI	M9-HI	M10-HI	×
	SET PT	16K	16K	×	250F	250F	250F	250F	×
CONTROL	LIMIT	M4 HI/LO	M6 HI/LO	×	×	×	×	×	×
	SET PT.	10.5/15K	10.5/15K	×	×	×	×	×	×
SIGNAL	CNDTR	M1	M2	×	×	×	×	×	×
	OUTPUT	0-5 VDC	0-5 VDC	×	×	×	×	×	×
SENSOR	N/A	030	030	030					

DATA SHEET TABLE 8.5 RS-31 DATA SYSTEMS

		LF CAV TEMP	AFT OIL PUMP TEMP	OIL CLR INLET TEMP	OIL CLR OUTLET TEMP				ENGINE TORQUE
MEAS. NO.		m9	m10	m11	m12	m13	m14	m15	m16
CODE		T-LE	T-AP	T-CI	T-CO				TQ-E
SENSOR	MFG	RC1282	→	→	→				
↓	P/N	-3S	-3C	-3C	-3C				(GFP)
	RANGE	0-300 F	→	→	→				0-1300 FT #
	TAP THRD	1/8 NPT	→	→	→				
	TAP DPTH	1.200	1" TEE	1 1/4 TEE	1" TEE				(GFP)
	LOCATION ↓	MIDDLE CASE	OIL LINE #10	OIL LINE #9	OIL LINE #6				LYCOMING GAS TURBINE
	DRAWING	026	071	071	071				
	ZONE	A14	SHT 1	SHT 1	SHT 1				
↓	VIEW	K	LO 7/8	OIL CLR INLET	LO 1				
CABLES	SENSOR	F Fe/CN	F →	F →	F →				(GFP)
	DISPLAY #	m9S	m10S	x	x				
	- FROM	M11	M12	x	x				
	CALIBR #	m9A	m10A	x	x				
	- FROM	M11	M12	x	x				
	RECORD #	m9F	m10F	m11F	m12F				
↓	- FROM	M11	M12	SENSOR	SENSOR				
SWITCHING	CALIB.	9-4B	9-9B	x	x				
↓	DISPLAY	9-4A	9-9A	x	x				
OUTPUT	DISPLAY	D4	D9	x	x				D21
	- VOLTS	0-2VDC	0-2VDC	x	x				
	RECORD	T5	x	T6	T7				
	- VOLTS	mV	x	mV	mV				
↓	ALM. LITE	L11M	L12M	x	x				
NOM. RTD VALUE		212°F	150 F	200 F	150F				1000 FT #
OBS. RED LINE		D4	D9	x	x				D21
ALARM	LIMIT	M11-HI	M12-HI	x	x				x
↓	SET PT	250 F	200 F	x	x				x
CONTROL	LIMIT	x	x	x	x				x
↓	SET PT.	x	x	x	x				x
SIGNAL	CNDTR	x	x	x	x				x
	OUTPUT	x	x	x	x				x

DATA SHEET TABLE 8.5 RS-31 DATA SYSTEMS

		ENGINE EXH. TEMP	ENGINE SPEED	ENGINE SPEED	RF CAV PRES	LF CAV PRES	FWD OIL PUMP PRES	AFT OIL PUMP PRES	GB OIL PUMP PRES
MEAS. NO.		m17	m18	m19	m20	m21	m22	m23	m24
CODE		T-EE	N-GT1	N-GT2	P-RC	P-LC	P-FP	P-AP	P-GP
SENSOR	MFG	(GFP)	→	→	HASTINGS		F	F	F
	P/N		→	→	DV-34	DV-34			
	RANGE	0-1000°C	0-120%	0-20K RPM	0-20 mm HG	0-20 mm HG	0-100 PSIG	0-500 PSIG	0-100 PSIG
	TAP THRD	(GFP)	→	→	1/8 NPT	→			
	TAP DPTH		→	→	1" TEE	1" TEE	1/2" TEE	1" TEE	
	LOCATION	LYCOMING GAS TURBINE			VAC LINE #1A	VAC LINE #1B	OIL LINE #11	OIL LINE #10	GEAR BOX
	DRAWING				071	071	071	071	ROK 1
	ZONE				SHT 1	SHT 1	SHT 1	SHT 1	SHT 1
	VIEW				VAC 5A	VAC 6A	LO 9/10	LO 7/8	LKG FWD
CABLES	SENSOR	GFP	GFP	GFP	R HASTINGS		F	F	F
	DISPLAY #				m20	m21	m22T	m23T	m24T
	-FROM				SENSOR	SENSOR	TAP	TAP	TAP
	CALIB #				x	x	x	x	x
	-FROM				x	x	x	x	x
	RECORD #				x	x	m22F	m23F	m24F
	-FROM				x	x	SENSOR	SENSOR	SENSOR
SWITCHING	CALIB.				x	x	x	x	x
	DISPLAY				x	x	x	x	x
OUTPUT	DISPLAY	D22	D23	D24	D8	D7	P4	P3	P6
	-VOLTS				}	}	(PSIG)	(PSIG)	(PSIG)
	RECORD				x	x	T10	T11	T12
	-VOLTS				x	x			
	ALM. LITE				L8C	L7C	x	x	x
NOM. RTD VALUE		800°C	70%	10.5-15K RPM	5 mm	5 mm	60 PSIG	500 PSIG	0-40 PSIG
OBS. RED LINE		D22	D23	D24	D8	D7	P4	P3	P6
ALARM	LIMIT	x	x	x	D8-HI	D7-HI	x	x	x
	SET PT	x	x	x	10 mm	10 mm	x	x	x
CONTROL	LIMIT	x	x	x	x	x	x	x	x
	SET PT.	x	x	x	x	x	x	x	x
SIGNAL	CNDTR	x	x	x	x	x	x	x	x
	OUTPUT	x	x	x	x	x	x	x	x

DATA SHEET TABLE 8.5 RS-31 DATA SYSTEMS

			RF GAP	LF GAP	RR GAP	LF GAP	RF VIBR	LF VIBR	RR VIBR
MEAS.	NO.	m25	m26	m27	m28	m29	m30	m31	m32
CODE			G-RF	G-LF	G-RR	G-LF	V-RF	V-LF	V-RR
SENSOR	MFG		ROCKETDYNE	→	→	→	ENDEVCO	→	→
	P/N		LE76-018-ER	→	→	→	2220C	→	→
	RANGE		.075-0"	.075-0"	.075-0"	.075-0"	0-20 G	0-20 G	0-20 G
	TAP THRD		7/8-14 UNF3A	→	→	→	.086-56	→	→
	TAP DPTH		THRU	→	→	→	.31"	→	→
	LOCATION		FWD CASE	MIDDLE CASE	MIDDLE CASE	AFT CASE	FWD CASE	FWD CASE	MIDDLE CASE
	DRAWING		027	026	026	025	027	027	026
	ZONE		B7	F12	G44	G8	D9(2)	D11(2)	D44/E46
	VIEW		BASIC	→	FF	BASIC	→	→	FF
CABLES	SENSOR		F 3/8 SH	F →	F →	F →	R ENDEVCO	→	→
	DISPLAY #		m26$	m27$	m28$	m29$	(m32$)	m31$	m32$
	- FROM		M13	M14	M15	M16	(M18)	M17	M18
	CALIBR #		m26A	m27A	m28A	m29A	(m32A)	m31A	m32A
	- FROM		M13	M14	M15	M16	(M18)	M17	M18
	RECORD #		M26F	m27F	m28F	m29F	m30F	m31F	m32F
	- FROM		M19	M20	M21	M22	M23	M24	M25
SWITCHING	CALIB.		9-11B	9-2B	9-11C	9-2C	(9-10B)	9-3B	9-10B
	DISPLAY		5-1A 9-11A	5-2A 9-2A	5-1B 9-11A	5-2B 9-2A	(9-10A)	9-3A	9-10A
OUTPUT	DISPLAY		D11	D2	(D11)	(D2)	(D10)	D3	D10
	- VOLTS		0-2 VDC	0-2 VDC	0-2 VDC	0-2 VDC	0-2 VDC	0-2 VDC	0-2 VDC
	RECORD		T13	T14	T15	T16	T17	T18	T19
	- VOLTS		0-1.88 VDC	0-1.88 VDC	0-1.88 VDC	0-1.88 VDC	0-2 VDC	0-2 VDC	0-2 VDC
	ALM. LITE		L13M	L14M	L15M	L16M	(L18M)	L17M	L18M
NOM. RTD	VALUE		.048"	.048"	.048"	.048"	0-1.5 G	0-1.5G	0-1.5G
OBS. RED	LINE		D11	D2	(D11)	(D2)	(D10)	D3	D10
ALARM	LIMIT		M13-HI	M14-HI	M15-HI	M16-HI	(M18-HI)	M17-HI	M18-HI
	SET PT		.055"	.055"	.055"	.055"	10G	10G	10G
CONTROL	LIMIT		x	x	x	x	x	x	x
	SET PT.		x	x	x	x	x	x	x
SIGNAL	CNDTR		M19	M20	M21	M22	M23	M24	M25
	OUTPUT		0-1.88 VDC	0-1.88 VDC	0-1.88 VDC	0-1.88 VDC	0-2 VDC	0-2 VDC	0-2 VDC

DATA SHEET TABLE 8.5 RS-31 DATA SYSTEMS

		LR VIBR		AFT OIL PR.SW	FWD OIL PR.SW.	BRK OIL PR.SW	CHIP DETECTOR	OIL ΔP FILTER SWITCHES	SUMP OIL LVL SWITCH
MEAS. NO.		M33	M34	M35	M36	M37	M38	M39	M40
CODE		V-LR		PS-AP	PS-FP	PS-BO	CD	ΔPS-F	ΔLS-S
SENSOR	MFG	ENDEVCO		CUSTOM	COMP.	SWITCHES	TEDECO	{ SEE -071- DWG }	EEPLS
↓	P/N	2220C		60795	60794	60794	JT402AK		-2050
	RANGE	0-20G		0-500 #	0-100 #	0-100 #	DNA	↓	
	TAP THRD	.086-56							1" NPT
	TAP DPTH	.31"							THRU
	LOCATION	AFT		OIL LINE	OIL LINE	OIL LINE	OIL LINE		AFT
	↓	CASE		#10	#11	#18	#5		CASE
	DRAWING	025		071	071	071	071	071	071
	ZONE	E48 H46		SHT 1	SHT 1	~~E8~~	SHT 1	SHT 1	SHT 1
↓	VIEW	FF		LO 7/8	LO 10/11	×	×	×	LO 3
CABLES	SENSOR	(R) ENDEVCO		(F)	(F)	(F)	(F) 3/C SH	(F) 3/C SH	(F) 3/C SH
↓	DISPLAY#	(M31B)		←(C11 & C12)→			C13	C21	C22
	-FROM	(M17)		← SENSORS & 5 VDC BUSS →					
	CALIBR#	(M31A)		×	×	×	×	×	×
	-FROM	M17		×	×	×	×	×	×
	RECORD#	M33F		×	×	×	×	×	×
↓	-FROM	M26		×	×	×	×	×	×
SWITCHING	CALIB.	(9-3B)		×	×	×	×	×	×
↓	DISPLAY	(9-3A)		×	×	×	×	×	×
OUTPUT	DISPLAY#	(B3)		×	×	×	×	×	×
↓	-VOLTS	0-2 VDC		×	×	×	×	×	×
	RECORD	T20		×	×	×	×	×	×
	- VOLTS	0-2 VDC		×	×	×	×	×	×
↓	ALM. LITE	(L17M)		L3C	L4C	L5C	L6C	L2C	L1C
NOM. RTD VALUE		0-1.5G		500 #G	60 #G	50 #G	CLEAN		
OBS. RED LINE		(D3)		×	×	×	×	×	×
ALARM	LIMIT	(M17-HI)		SENSOR	→	→	→	→	→
↓	SET PT	10G		425 #G	40 #G	40 #G	SHORTED		
CONTROL	LIMIT	×		×	×	×	×	×	×
↓	SET PT.	×		×	×	×	×	×	×
SIGNAL	CNDTR	M26		×	×	×	×	×	×
	OUTPUT	0-2 VDC		×	×	×	×	×	×

DATA SHEET TABLE 8.5 RS-31 DATA SYSTEMS

		LF BRK TEMP	LR BRK TEMP	RR BRK TEMP	RF BRK TEMP	LF BRK VIBR	LR BRK VIBR	RF BRK VIBR	RR BRK VIBR
MEAS.	NO.	m41	m42	m43	m44	m45	m46	m47	m48
CODE		T-BLF	T-BLR	T-BRR	T-BRF	V-BLF	V-BLR	V-BRF	V-BRR
SENSOR	MFG	RC-1282	→	→	→	F	F	F	F
↓	P/N	-63	-63	-63	-63				
	RANGE	0-300F	→	→	→	0-20G	→	→	→
	TAP THRD	1/8 NPT	→	→	→	.086-56	→	→	→
	TAP DPTH	2.044	2.089	1.918	2.089	.31	→	→	→
	LOCATION	LH	→	RH	→	LH	→	RH	→
	↓	BRAKE	→	BRAKE	→	BRAKE	→	BRAKE	→
	DRAWING	208	→	206	→	208	→	206	→
	ZONE	E15	C15	B10	G8	A8	A8	C5	C5
	VIEW	EE	FF	DD	AA	JJ	KK	FF	GG
CABLES,	SENSOR	F Fe/Cn	F →	F →	F →	F	F →	F →	F →
↓	DISPLAY #	×	×	×	×	×	×	×	×
	- FROM	×	×	×	×	×	×	×	×
	CALIBR #	×	×	×	×	×	×	×	×
	- FROM	×	×	×	×	×	×	×	×
	RECORD #	m41F	m42F	m43F	m44F	m45F	m46F	m47F	m48F
	- FROM	SENSOR	→	→	→	FAC SIGNAL CONDITIONER	→	→	→
SWITCHING,	CALIB.	×	×	×	×	×	×	×	×
↓	DISPLAY	×	×	×	×	×	×	×	×
OUTPUT	DISPLAY	MP9	MP10	MP11	MP12	×	×	×	×
↓	- VOLTS	mv	mv	mv	mv	×	×	×	×
	RECORD	MP9	MP10	MP11	MP12	T21	T22	T23	T24
	- VOLTS	mv	mv	mv	mv	VDC	VDC	VDC	VDC
	ALM. LITE	×	×	×	×	×	×	×	×
NOM. RTD	VALUE	150F	150 F	150 F	150 F	0-1.5 G	0-1.5 G	0-1.5 G	0-1.5 G
OBS. RED	LINE	MP9	MP10	MP11	MP12	×	×	×	×
ALARM	LIMIT	×	×	×	×	×	×	×	×
↓	SET PT	×	×	×	×	×	×	×	×
CONTROL	LIMIT	×	×	×	×	×	×	×	×
↓	SET PT.	×	×	×	×	×	×	×	×
SIGNAL	CNDTR	×	×	×	×	F	F	F	F
	OUTPUT	×	×	×	×				

DATA SHEET TABLE 8.5 RS-31 DATA SYSTEMS

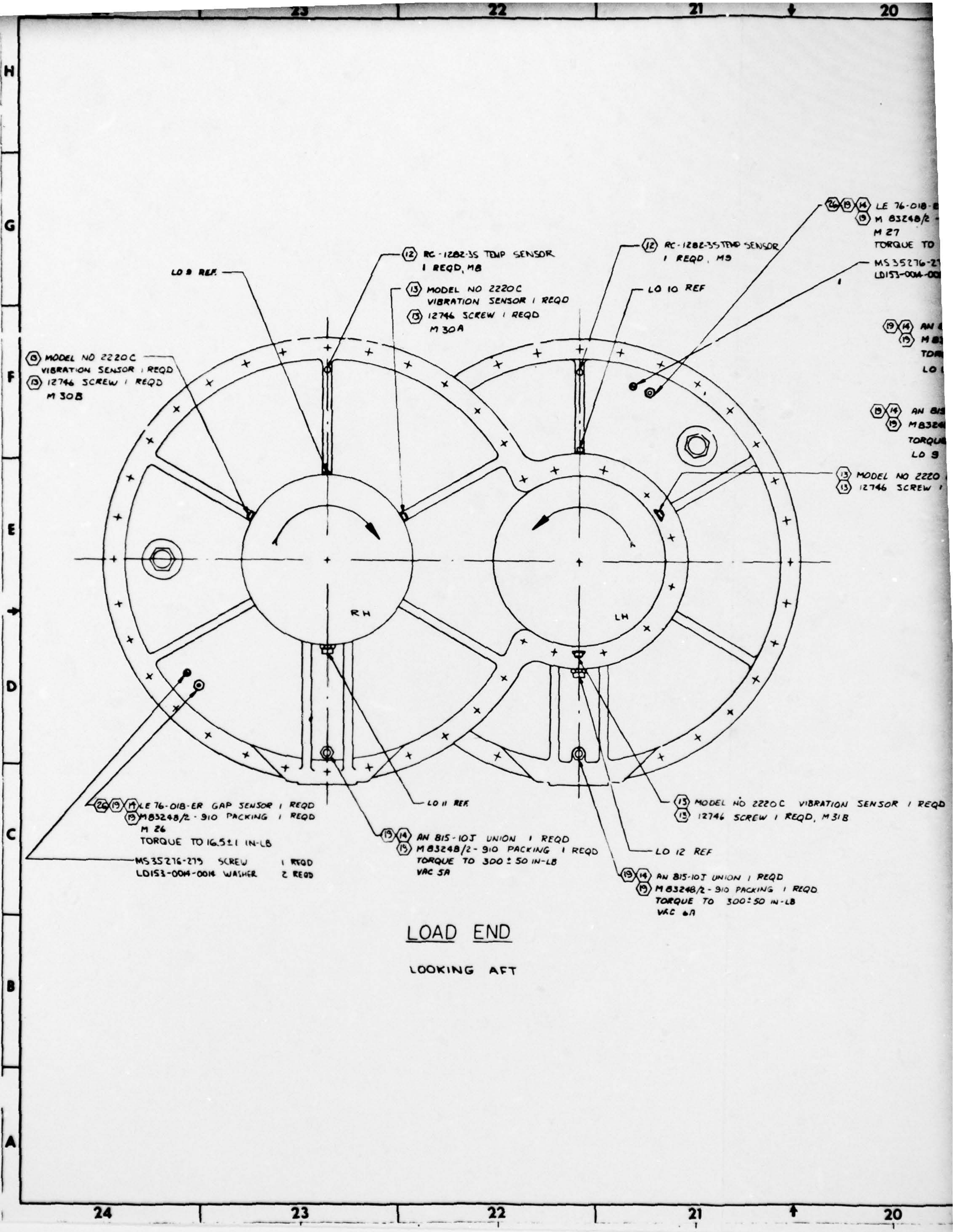
		FWD OIL FLOW	AFT OIL FLOW	RR BRK PRES	RF BRK PRES	LR BRK PRES	LF BRK PRES	BRAKE VACUUM	BRAKE INC. AIR PRES
MEAS.	NO.	M49	M50	M51	M52	M53	M54	M55	M56
CODE		Q-FP	Q-AP	P-BRR	P-BRF	P-BLR	P-BLF	P-BV	P-BA
SENSOR	MFG	F	F	F	F	F	F	F	F
	P/N								
	RANGE	0-2.5 GPM	0-7.5 GPM	0-30 PSIA	→	→	→	0-15 PSIA	0-120 PSIG
	TAP THRD	½" TUBE	1" TUBE	¼ TUBE	→	→	→		
	TAP DPTH	x	x	STD	→	→	→		
	LOCATION	OIL LINE	OIL LINE	RH	→	LH	→	VAC LINE	AIR LINE
↓		#11	#10	BRAKE	→	BRAKE	→	#16	#14
	DRAWING	071	071	206	→	208	→	071	071
	ZONE	x	x	D15	E8	F10	B9	D10	D12
	VIEW	SHT 1	SHT 1	BB	AA	AA	DD	SHT 2	SHT 2
CABLES,	SENSOR	F	F	F	F	F	F	F	F
	DISPLAY #	x	x	x	x	x	x	M55S	M56S
	- FROM	x	x	x	x	x	x	PM17B	PM17A
	CALIBR #	x	x	x	x	x	x		
	- FROM	x	x	x	x	x	x		
	RECORD #	M49F	M50F	M51F	M52F	M53F	M54F	M55F	M56F
	- FROM								
SWITCHING,	CALIB.	x	x	x	x	x	x	12-A	13-A
↓	DISPLAY	x	x	x	x	x	x	12-B	13-B
OUTPUT	DISPLAY	B7	B8	x	x	x	x	P1	P2
	- VOLTS			x	x	x	x	1-5 VDC	1-5 VDC
	RECORD	B7	B8	T25	T26	T27	T28	T9	T8
	- VOLTS								
↓	ALM. LITE	x	x	x	x	x	x	x	x
NOM. RTD VALUE		2.5 GPM	7.5 GPM	5 PSIA	5 PSIA	5 PSIA	5 PSIA	2 PSIA	120 PSIG
OBS. RED LINE		B7	B8	x	x	x	x	P1	P2
ALARM	LIMIT	x	x	x	x	x	x	x	x
↓	SET PT	x	x	x	x	x	x	x	x
CONTROL	LIMIT	x	x	x	x	x	x	PM-7	PM-8
↓	SET PT.	x	x	x	x	x	x	2 PSIA	120 PSIG
SIGNAL	CNDTR	F						PM-7	PM-8
	OUTPUT	F						10-50 mA	10-50 mA

DATA SHEET TABLE 8.5 RS-31 DATA SYSTEMS

		BRAKE OIL INL PRES		BRAKE OIL FLOW	GEAR BOX OIL FLOW	GB 1F BRG TEMP	GB 1A BRG TEMP	GB 2F BRG TEMP	GB 2A BRG TEMP
MEAS. NO.		m57	m58	m59	m60	m61	m62	m63	m64
CODE		P-B0		Q-B0	Q-G0	T-G1F	T-G1A	T-G2F	T-G2A
SENSOR	MFG	F		F	F	RC 1282	→	→	→
	P/N					-6$	-6$	-6$	-6$
	RANGE	0-400 PSIG		0-16 GPM	0-9 GPM	0-300 F	→	→	→
	TAP THRD			1" LINE	1½" LINE	⅛ NPT	→	→	→
	TAP DPTH			}	}	4.000	4.000	4.250	4.250
	LOCATION	OIL LINE		OIL LINE	OIL LINE	(GEAR	→	→	→
	↓	#18		#18	#12	BOX)	→	→	→
	DRAWING	071		071	071	(030 & ROK 8)		→	→
	ZONE	E8		E8	x	C11	C11	C11	C11
	VIEW	SHT 2		SHT 2	SHT 1	(LKG	FWD)	→	→
CABLES,	SENSOR	F		F	F	F Fe/Cn	F →	F →	F →
	DISPLAY #	M57T		x	x	x	x	x	x
	-FROM	TAP		x	x	x	x	x	x
	CALIBR #	x		x	x	x	x	x	x
	-FROM	x		x	x	x	x	x	x
	RECORD #	x		M59F	M60F	M61F	M62F	M63F	M64F
	-FROM	x		SENSOR	SENSOR	SENSOR	SENSOR	SENSOR	SENSOR
SWITCHING,	CALIB.	x		x	x	x	x	x	x
↓	DISPLAY	x		x	x	x	x	x	x
OUTPUT	DISPLAY	P5		B5	B6	MP1	MP2	MP3	MP4
	-VOLTS	[PSIG]				mv	mv	mv	mv
	RECORD	x		B5	B6	MP1	MP2	MP3	MP4
	-VOLTS	x				mv	mv	mv	mv
	ALM. LITE	x		x	x	x	x	x	x
NOM. RTD VALUE		50 & 400 PSIG		6.3 & 15.7 GPM	0-9 GPM	200 F	200 F	200 F	200 F
OBS. RED LINE		P5		B5	B6	MP1	MP2	MP3	MP4
ALARM	LIMIT	x		x	x	x	x	x	x
↓	SET PT	x		x	x	x	x	x	x
CONTROL	LIMIT	x		x	x	x	x	x	x
↓	SET PT.	x		x	x	x	x	x	x
SIGNAL	CNDTR	x				x	x	x	x
	OUTPUT	x				x	x	x	x

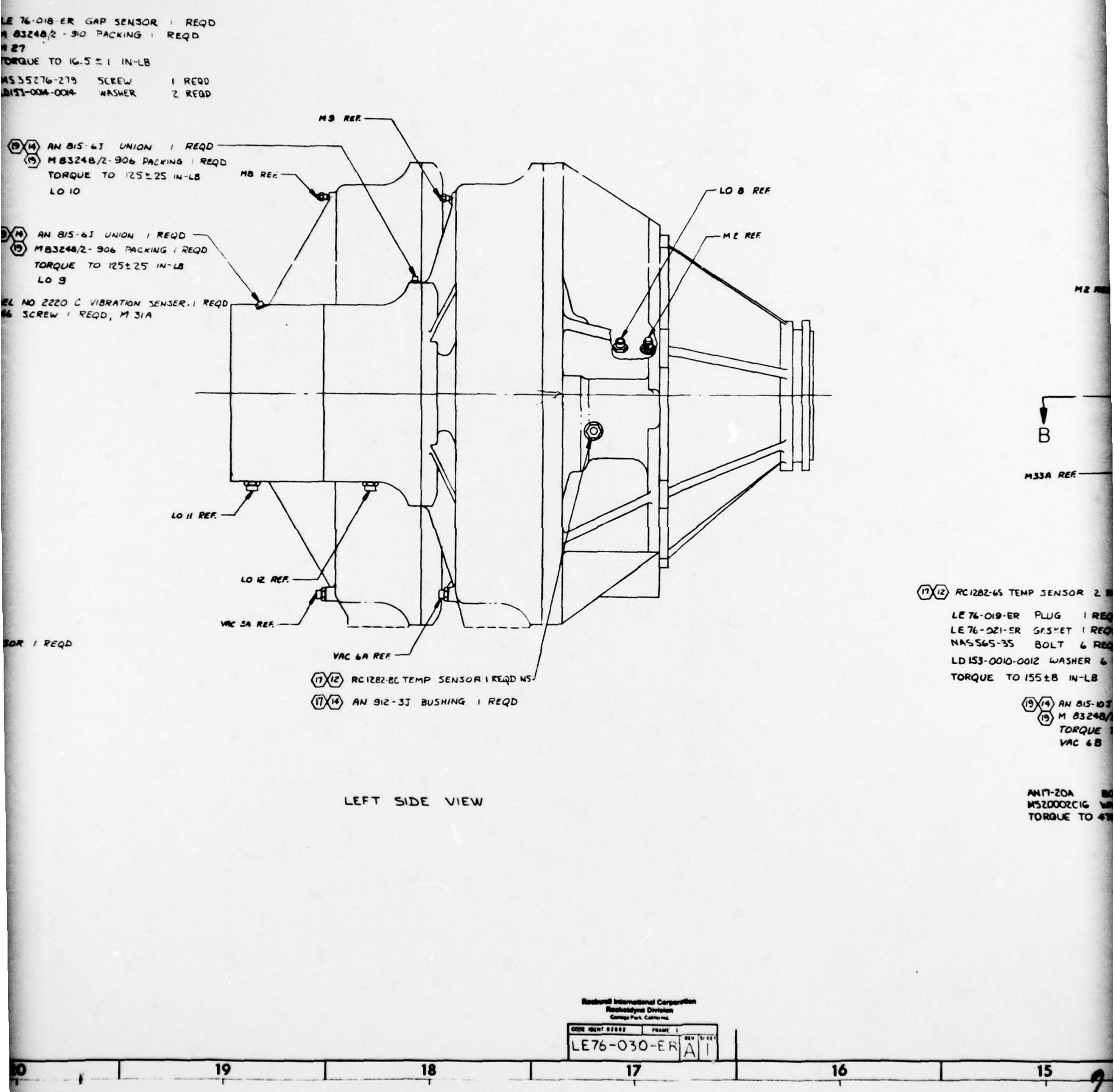
DATA SHEET TABLE 8.5 RS-31 DATA SYSTEMS

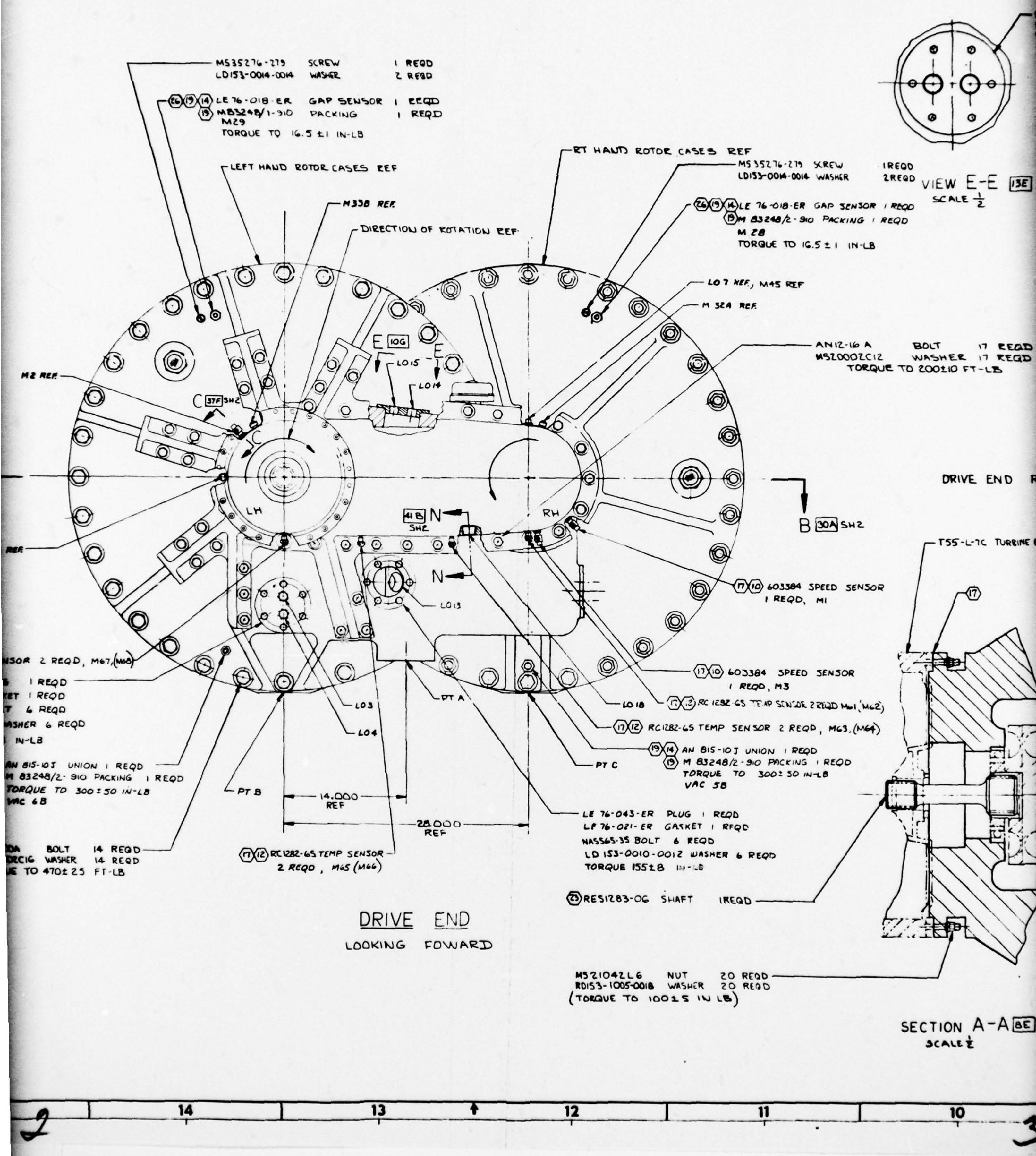
		GB 3F TEMP	GB 3A TEMP	GB 4F TEMP	GB 4A TEMP	GB OIL OUT TEMP	GB #1 CASE TEMP	GB #2 CASE TEMP	
MEAS.	NO.	m65	m66	m67	m68	m69	m70	m71	m72
CODE		T-G3F	T-G3A	T-G4F	T-G4A	T-GOO	T-GC1	T-GC2	
SENSOR	MFG	RC-1282	→	→	→	→	F	F	
↓	P/N	-65	-65	-65	-65	-3C			
	RANGE	0-300F	→	→	→	0-200F	→	→	
	TAP THRD	1/8 NPT	→	→	→	→	NONE		
	TAP DPTH	4.250	4.250	3.500	3.500	THRU	"	"	
	LOCATION	(GEAR	→	→	→	→	→	→	
	↓	BOX)	→	→	→	→	→	→	
	DRAWING	(030 #	RAK 8)	→	→	→	→	→	
	ZONE	B13	B13	D15	D15				
	VIEW	(LKG FWD)	→	→	→				
CABLES,	SENSOR	F Fe/Cn	F →	F →	F →	F →	F	F	
↓	DISPLAY	x	x	x	x	x	x	x	
	- FROM	x	x	x	x	x	x	x	
	CALIBR	x	x	x	x	x	x	x	
	- FROM	x	x	x	x	x	x	x	
	RECORD	M65F	M66F	m67F	m68F	m69F	m70F	m71F	
	- FROM	SENSOR	→	→	→	→	→	→	
SWITCHING,	CALIB.	x	x	x	x	x	x	x	
↓	DISPLAY	x	x	x	x	x	x	x	
OUTPUT	DISPLAY	MP5	MP6	MP7	MP8	ALT	ALT	ALT	
↓	- VOLTS	mV	mV	mV	mV	mV	mV	mV	
	RECORD	MP5	MP6	MP7	MP8	ALT	ALT	ALT	
	- VOLTS	mV	mV	mV	mV	mV	mV	mV	
	ALM. LITE	x	x	x	x	x	x	x	
NOM. RTD	VALUE	200 F	200 F	200 F	200 F	200 F	200 F	200 F	
OBS. RED	LINE	MP5	MP6	MP7	MP8	ALT	ALT	ALT	
ALARM	LIMIT	x	x	x	x	x	x	x	
↓	SET PT	x	x	x	x	x	x	x	
CONTROL	LIMIT	x	x	x	x	x	x	x	
↓	SET PT.	x	x	x	x	x	x	x	
SIGNAL	CNDTR	x	x	x	x	x	x	x	
	OUTPUT	x	x	x	x	x	x	x	

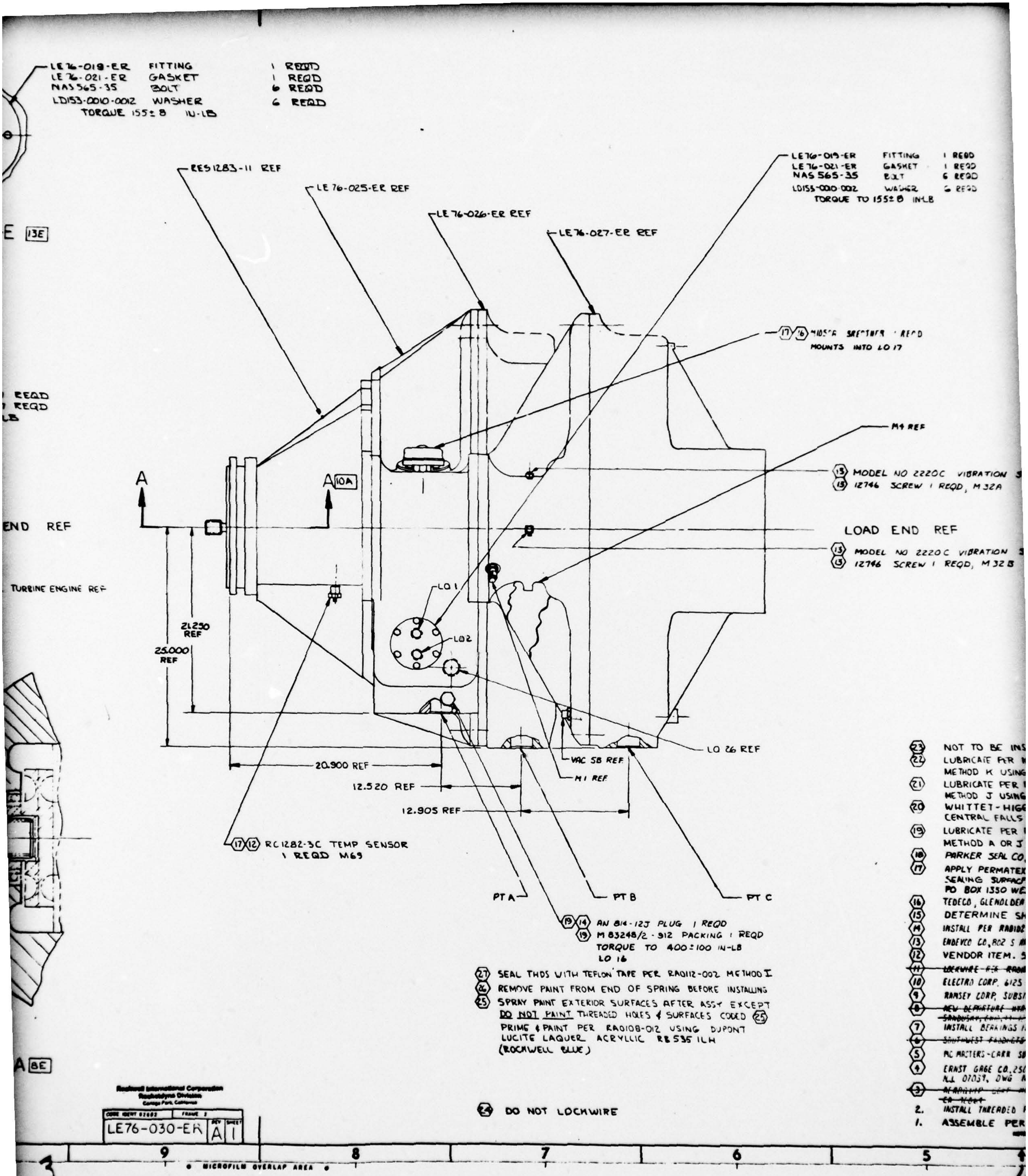


LOAD END

LOOKING AFT


LE 76-018-ER GAP SENSOR 1 REQD
M 83248/2 - 910 PACKING 1 REQD
M 27
TORQUE TO 16.5 ± 1 IN-LB
MS35276-279 SCREW 1 REQD
LD153-0014-0014 WASHER 2 REQD
M 9 REF.
(19)(14) AN 815-6J UNION 1 REQD
(19) M 83248/2-906 PACKING 1 REQD
TORQUE TO 125±25 IN-LB
LO 10
M8 REF.
(14) AN 815-6J UNION 1 REQD
(19) M 83248/2-906 PACKING 1 REQD
TORQUE TO 125±25 IN-LB
LO 9
EL NO 2220 C VIBRATION SENSER. 1 REQD
6 SCREW 1 REQD, M 31A
LO 8 REF.
M 2 REF.
M2 REF
B
M33A REF.
LO 11 REF.
LO 12 REF.
VAC 5A REF.
VAC 6A REF.
SOR 1 REQD
(17)(12) RC1282-8C TEMP SENSOR 1 REQD M5
(17)(14) AN 912-3J BUSHING 1 REQD
(17)(12) RC1282-6S TEMP SENSOR 2
LE 76-019-ER PLUG 1 REQ
LE 76-021-ER GASKET 1 REQ
NAS565-35 BOLT 6 REQ
LD 153-0010-0012 WASHER 6
TORQUE TO 155±8 IN-LB
(19)(14) AN 815-10J
(19) M 83248/
TORQUE
VAC 6B
AN17-20A
MS20002C16
TORQUE TO 47
LEFT SIDE VIEW
Rockwell International Corporation
Rocketdyne Division
Canoga Park, California
CODE IDENT 02602
FRAME 1
LE76-030-ER
REV A
SHEET 1
20
19
18
17
16
15


MS35276-279 SCREW 1 REQD
LD153-0014-0014 WASHER 2 REQD
26 19 14 LE 76-018-ER GAP SENSOR 1 REQD
19 M83248/1-910 PACKING 1 REQD
M29
TORQUE TO 16.5 ±1 IN-LB
LEFT HAND ROTOR CASES REF
M33B REF
DIRECTION OF ROTATION REF
RT HAND ROTOR CASES REF
MS35276-279 SCREW 1 REQD
LD153-0014-0014 WASHER 2 REQD
26 19 14 LE 76-018-ER GAP SENSOR 1 REQD
19 M 83248/2-910 PACKING 1 REQD
M 28
TORQUE TO 16.5 ± 1 IN-LB
VIEW E-E 13E
SCALE 1/2
LO 7 REF, M45 REF
M 32A REF
AN12-16 A BOLT 17 REQD
MS20002C12 WASHER 17 REQD
TORQUE TO 200±10 FT-LB
M2 REF
C 37F SH2
E 10G
LO15
LO14
LH
RH
41B SH2
N
B 30A SH2
DRIVE END
T55-L-7C TURBINE
17
17 10 603384 SPEED SENSOR 1 REQD, M1
LO13
17 10 603384 SPEED SENSOR 1 REQD, M3
LO18
17 12 RC 1282-6S TEMP SENSOR 2 REQD M61 (M62)
17 12 RC1282-6S TEMP SENSOR 2 REQD, M63, (M64)
19 14 AN 815-10J UNION 1 REQD
19 M 83248/2-910 PACKING 1 REQD
TORQUE TO 300±50 IN-LB
VAC 5B
PT A
PT B
PT C
LO3
LO4
NSOR 2 REQD, M67, (M68)
1 REQD
1 REQD
6 REQD
ASHER 6 REQD
IN-LB
AN 815-10J UNION 1 REQD
M 83248/2-910 PACKING 1 REQD
TORQUE TO 300±50 IN-LB
VAC 6B
14.000 REF
28.000 REF
BOLT 14 REQD
WASHER 14 REQD
TO 470±25 FT-LB
17 12 RC1282-6S TEMP SENSOR 2 REQD, M65 (M66)
LE 76-043-ER PLUG 1 REQD
LP 76-021-ER GASKET 1 REQD
NAS565-35 BOLT 6 REQD
LD 153-0010-0012 WASHER 6 REQD
TORQUE 155±8 IN-LB
23 RES1283-06 SHAFT 1 REQD
DRIVE END
LOOKING FOWARD
MS21042L6 NUT 20 REQD
RD153-1005-0018 WASHER 20 REQD
(TORQUE TO 100±5 IN LB)
SECTION A-A 8E
SCALE 1/2
2
14
13
12
11
10
3



LE76-018-ER FITTING 1 REQD
LE76-021-ER GASKET 1 REQD
NAS565-35 BOLT 6 REQD
LD153-0010-0012 WASHER 6 REQD
TORQUE 155±8 IN-LB
RES1283-11 REF
LE76-025-ER REF
LE76-026-ER REF
LE76-027-ER REF
LE76-019-ER FITTING 1 REQD
LE76-021-ER GASKET 1 REQD
NAS565-35 BOLT 6 REQD
LD153-0010-0012 WASHER 6 REQD
TORQUE TO 155±8 IN-LB
E 13E
M4 REF
A
A 10A
END REF
LOAD END REF
TURBINE ENGINE REF
(15) MODEL NO 2220C VIBRATION
(13) 12746 SCREW 1 REQD, M32A
(13) MODEL NO 2220C VIBRATION
(13) 12746 SCREW 1 REQD, M32B
LO1
LO2
21.250 REF
25.000 REF
20.900 REF
12.520 REF
12.905 REF
LO 26 REF
VAC 58 REF
M1 REF
(17)(12) RC1282-3C TEMP SENSOR 1 REQD M69
PT A
PT B
PT C
(19)(14) AN 814-12J PLUG 1 REQD
(19) M83248/2-912 PACKING 1 REQD
TORQUE TO 400±100 IN-LB
LO 16
(27) SEAL THDS WITH TEFLON TAPE PER RA0112-002 METHOD I
(26) REMOVE PAINT FROM END OF SPRING BEFORE INSTALLING
(25) SPRAY PAINT EXTERIOR SURFACES AFTER ASSY EXCEPT DO NOT PAINT THREADED HOLES & SURFACES CODED (25)
PRIME & PAINT PER RA0108-012 USING DUPONT LUCITE LAQUER ACRYLLIC RB535 ILH (ROCKWELL BLUE)
(24) DO NOT LOCKWIRE
A 8E
(23) NOT TO BE INS
(22) LUBRICATE PER METHOD K USING
(21) LUBRICATE PER METHOD J USING
(20) WHITTET-HIGG CENTRAL FALLS
(19) LUBRICATE PER METHOD A OR J
(18) PARKER SEAL CO,
(17) APPLY PERMATEX SEALING SURFACE PO BOX 1350 WE
(16) TEDECO, GLENOLDEN
(15) DETERMINE SH
(14) INSTALL PER RA0102
(13) ENDEVCO CO, 802 S
(12) VENDOR ITEM.
(11) ~~LOCKWIRE PER RA0~~
(10) ELECTRO CORP. 6125
(9) RAMSEY CORP, SUBSI
(8) ~~NEW DEPARTURE~~
(7) INSTALL BEARINGS
(6) ~~SOUTHWEST PRODUCTS~~
(5) MC MASTERS-CARR
(4) ERNST GAGE CO, N.J. 07059, DWG
(3) ~~BEARING~~
2. INSTALL THREADED
1. ASSEMBLE PER
Rockwell International Corporation
Rocketdyne Division
Canoga Park, California
LE76-030-ER A 1
9 8 7 6 5
MICROFILM OVERLAP AREA
3
4

REVISIONS				
ZONE	LTR	DESCRIPTION	DISPN	DATE / APPROVED
		1. MAY BE REWORKED 2. CANNOT BE REWORKED 3. RECORD CHANGE 4. NOW SHOP PRACTICE 5. PARTS MADE OK		
	A	1) REDESIGN & REDRAWN.	3	
5	B	1) IN NOTE 22 FS 1281 WAS DC-55	3	
F/D		2) NAS 565-35 WAS AN 5-7A AT 4 LOCATIONS	3	
11		3) MS 20002 C12 WAS MS 20002 C16	3	
F/D		4) 18 M83248/2-910 WAS 19 M83248/1-910 AT 10 LOCATIONS	3	
12		5) ADDED TORQUE 100±5 IN LBS TO MS 21042L6 NUT 20 REQD	3	
8		6) ADDED 17 12 RC1282-3C 1 REQD	3	
4 F/D		7) ADDED NOTE 26 & ADDED 26 TO F/D 4 PLACES	3	
F/D		8) ADDED MS35276-279 1 REQD LD151-0014-0014 2 REQD (4 LOCATIONS)	3	
7		9) REVISED NOTE 25	3	
7		10) ADDED NOTE 27	3	

SENSOR 1 REQD

SENSOR 1 REQD

APPENDIX - B

INSTALLED DURING SHIPMENT OF ASSY
R KA0112-002 FOR SPECIAL CONDITIONS
SING FS1281 LUBRICANT
ER RA0112-002 FOR SPECIAL CONDITIONS
ING MIL-L-23699 OIL
IGGINS COMPANY
LS, R.I. 02863
R RA0112-002 FOR SPECIAL CONDITIONS
R J USING RB0140-012 LUBRICANT
CO, 10567 JEFFERSON BLVD CULVER CITY, CA
TEX # 3 TO METAL-TO-METAL
ACES & THREADS. PERMATEX CO.,
WEST PALM BEACH, FL 33402
DEN, PA
SHIM THICKNESS AT ASSY
0102-005
S ARROYO PARKWAY, PASADENA, CA
. SEE SOURCE CONTROL DOCUMENT
~~0101-002~~
25 W HOWARD ST, CHICAGO, ILL
BSIDIARY OF TRW, ST. LOUIS, MO
~~[illegible] BEARINGS, DIVISION OF GENERAL MOTORS,~~
~~[illegible] STANDARD LOCKWASHER,~~
S PER RA0210-565
~~[illegible]~~
SUPPLY CO. LOS ANGELES, CA 90054
250 S. LIVINGSTON AVE, LIVINGSTON,
NO. 665, CATALOG 10-B
~~[illegible] LOS ANGELES~~
FASTENERS PER RA0101-002
ER RC1288
[illegible] UNLESS OTHERWISE SPECIFIED

NEXT ASSY	USED ON
APPLICATION	

HEAT TREAT: NONE

FINISH: 25

MATL

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES AND APPLY PRIOR TO FINISH
MACH. SURF. ROUGHNESS
TOLERANCES ON ANGLES ± 0° 30'
DECIMALS .XX ± .03 .XXX ± .010
HOLES NOTED "DRILL"
DO NOT SCALE PRINT

DAN[illegible]-75-C-0[illegible]B
DWN [illegible] DATE
CHK DATE
DSGN
MATL
STRUCT
DESIGN ACTIVITY APVD H. H. ROSS DATE 6-3-76

Rockwell International Corporation
Rocketdyne Division
Canoga Park, California

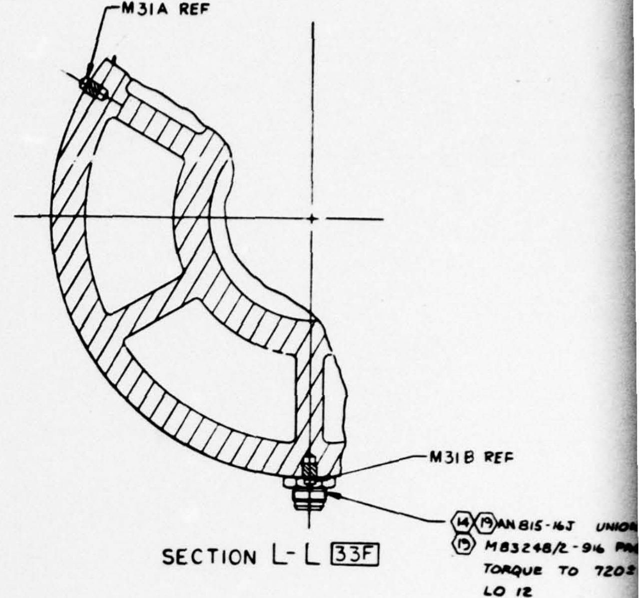
FLYWHEEL ASSY

SIZE J | CODE IDENT NO 02602 | DRAWING NO LE 76-030-[illegible]R

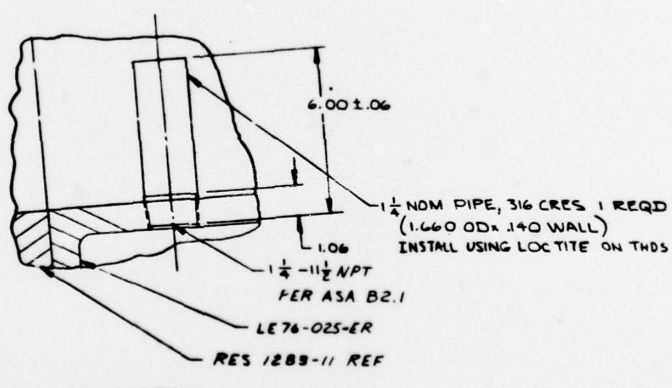
SCALE 1/4 SHEET [illegible]

H
G
F
E
D
C
B
A

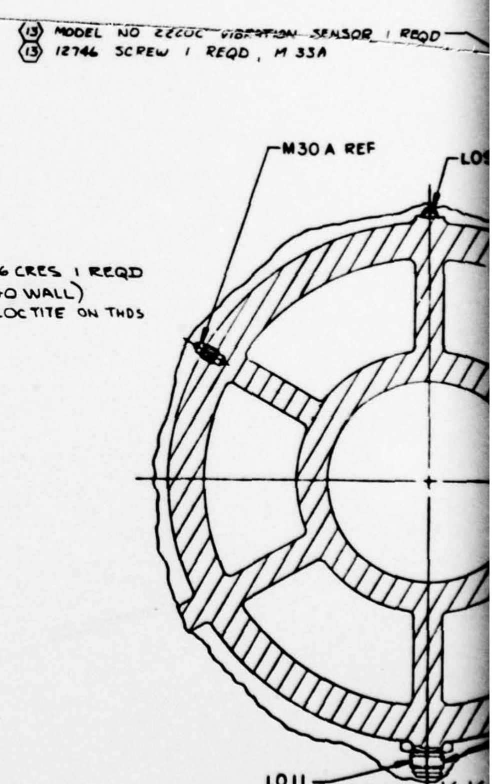
48 47 46 45 44



SECTION L-L 33F



SECTION N-N 12E SH1
LO 18 REF



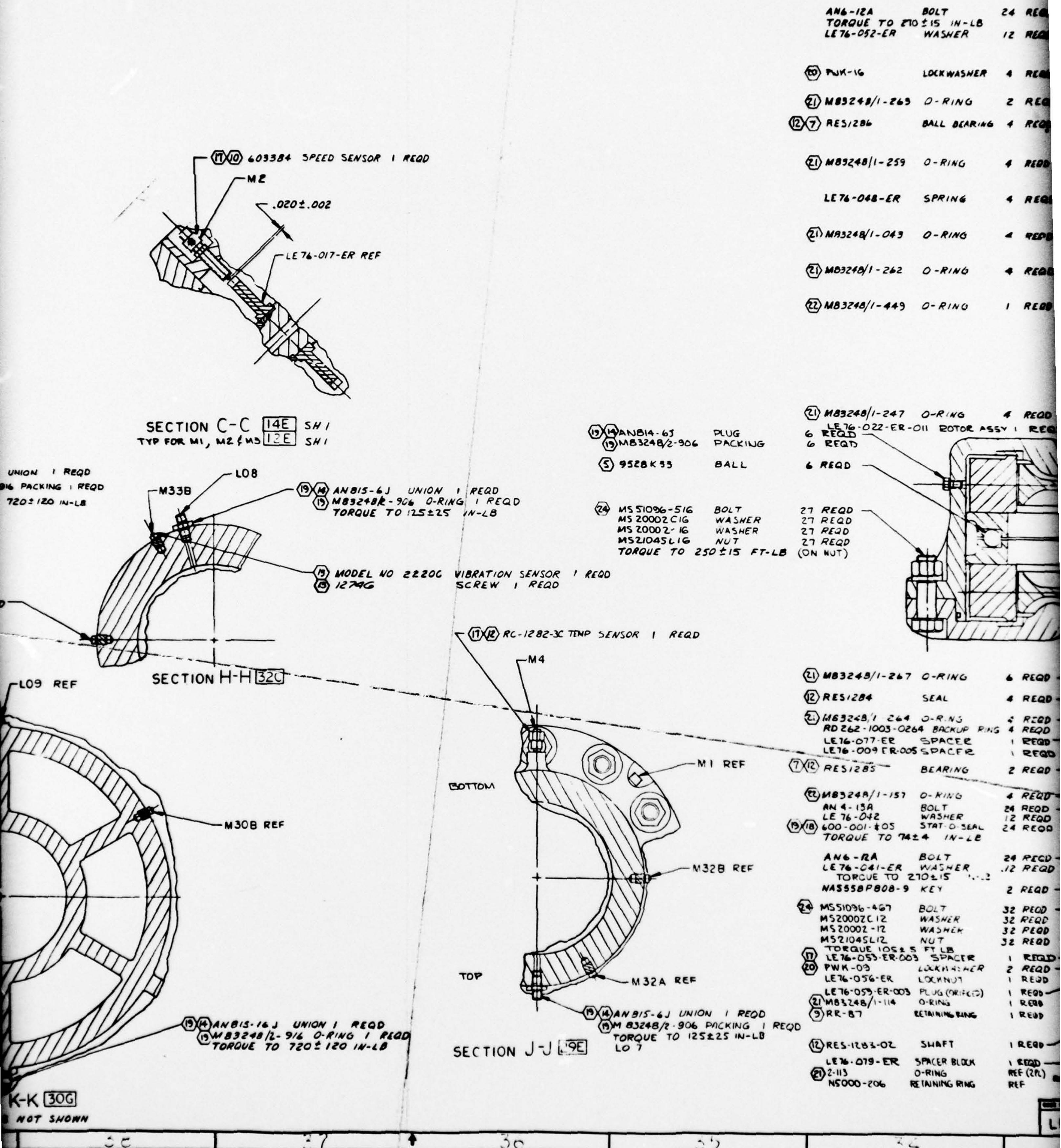
SECTION K-K
INTERNAL DETAILS NOT

Rockwell International Corporation
Rocketdyne Division
Canoga Park, California

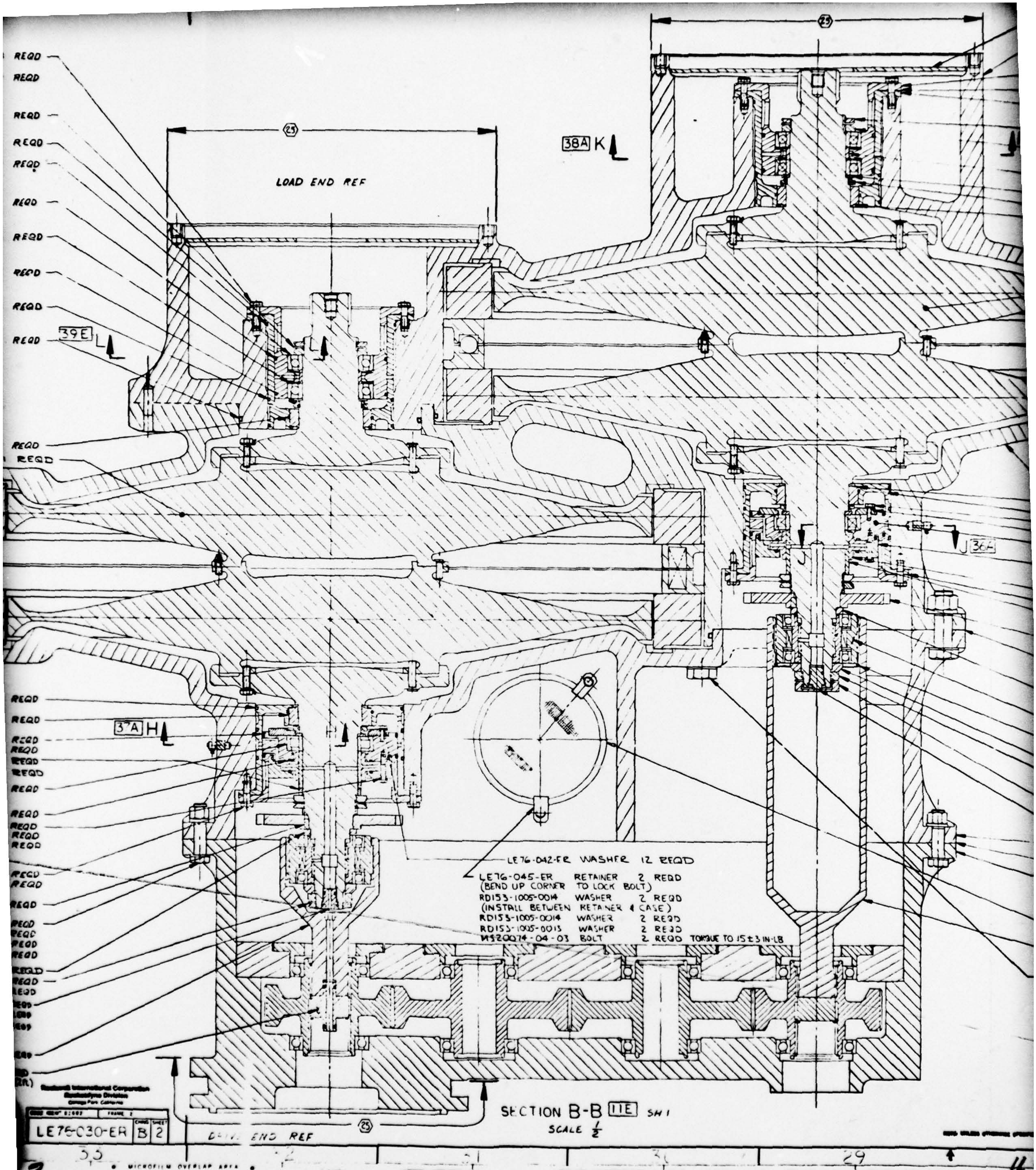
CODE IDENT 02602	FRAME 1		
LE76-030-ER		B	2

43 42 41 40 39

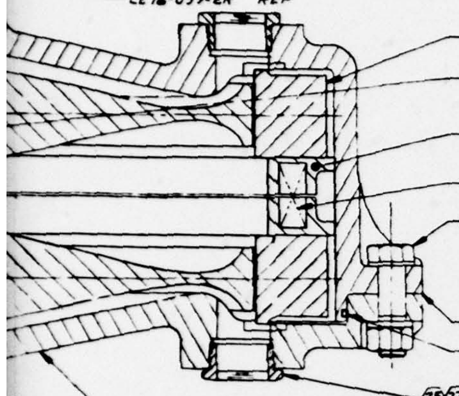
• MICROFILM OVERLAP AREA •



AN6-12A BOLT 24 REQ
TORQUE TO 270 ± 15 IN-LB
LE76-052-ER WASHER 12 REQ
(20) PWK-16 LOCKWASHER 4 REQ
(21) M83248/1-263 O-RING 2 REQ
(12)(7) RES1286 BALL BEARING 4 REQ
(21) M83248/1-259 O-RING 4 REQD
LE76-048-ER SPRING 4 REQ
(21) M83248/1-043 O-RING 4 REQ
(21) M83248/1-262 O-RING 4 REQ
(22) M83248/1-449 O-RING 1 REQ
(17)(10) 603384 SPEED SENSOR 1 REQD
M2
.020 ± .002
LE76-017-ER REF
SECTION C-C 14E SH 1
TYP FOR M1, M2 & M3 12E SH 1
(21) M83248/1-247 O-RING 4 REQD
LE76-022-ER-011 ROTOR ASSY 1 REQ
(19)(14) AN814-6J PLUG 6 REQD
(19) M83248/2-906 PACKING 6 REQD
(5) 9528K33 BALL 6 REQD
UNION 1 REQD
-916 PACKING 1 REQD
720 ± 120 IN-LB
L08
M33B
(19)(14) AN815-6J UNION 1 REQD
(19) M83248/2-906 O-RING 1 REQD
TORQUE TO 125 ± 25 IN-LB
(24) MS51096-516 BOLT 27 REQD
MS20002C16 WASHER 27 REQD
MS20002-16 WASHER 27 REQD
MS21045L16 NUT 27 REQD
TORQUE TO 250 ± 15 FT-LB (ON NUT)
(13) MODEL NO 2220C VIBRATION SENSOR 1 REQD
(13) 12796 SCREW 1 REQD
(17)(12) RC-1282-3C TEMP SENSOR 1 REQD
M4
SECTION H-H 32C
L09 REF
(21) M83248/1-267 O-RING 6 REQD
(12) RES1284 SEAL 4 REQD
(21) M83248/1-264 O-RING 4 REQD
RD262-1003-0264 BACKUP RING 4 REQD
LE76-077-ER SPACER 1 REQD
LE76-009-ER-005 SPACER 1 REQD
M1 REF
(7)(12) RES1285 BEARING 2 REQD
BOTTOM
M30B REF
(22) M83248/1-157 O-RING 4 REQD
AN4-13A BOLT 24 REQD
LE76-042 WASHER 12 REQD
(19)(18) 600-001-405 STAT-O-SEAL 24 REQD
TORQUE TO 74 ± 4 IN-LB
AN6-12A BOLT 24 REQD
LE76-041-ER WASHER 12 REQD
TORQUE TO 270 ± 15
NAS558P808-9 KEY 2 REQD
M32B REF
(24) MS51096-467 BOLT 32 REQD
MS20002C12 WASHER 32 REQD
MS20002-12 WASHER 32 REQD
MS21045L12 NUT 32 REQD
TORQUE 105 ± 5 FT LB
(17) LE76-053-ER-003 SPACER 1 REQD
(20) PWK-08 LOCKWASHER 2 REQD
LE76-056-ER LOCKNUT 1 REQD
LE76-059-ER-003 PLUG (ORIFICED) 1 REQD
(21) M83248/1-114 O-RING 1 REQD
(9) RR-87 RETAINING RING 1 REQD
TOP
M32A REF
(19)(14) AN815-16J UNION 1 REQD
(19) M83248/2-916 O-RING 1 REQD
TORQUE TO 720 ± 120 IN-LB
(19)(14) AN815-6J UNION 1 REQD
(19) M83248/2-906 PACKING 1 REQD
TORQUE TO 125 ± 25 IN-LB
L07
SECTION J-J 29E
(12) RES-1283-02 SHAFT 1 REQD
LE76-079-ER SPACER BLOCK 1 REQD
(21) 2-113 O-RING REF (2PL)
N5000-206 RETAINING RING REF
K-K 30G
NOT SHOWN
38
37
36
35
34


LOAD END REF
38A K
39E L
37A H
J 36A
LE76-042-ER WASHER 12 REQD
LE76-045-ER RETAINER 2 REQD
(BEND UP CORNER TO LOCK BOLT)
RD153-1005-0014 WASHER 2 REQD
(INSTALL BETWEEN RETAINER & CASE)
RD153-1005-0014 WASHER 2 REQD
RD153-1005-0013 WASHER 2 REQD
MS20074-04-03 BOLT 2 REQD TORQUE TO 15±3 IN-LB
SECTION B-B 11E SH 1
SCALE 1/2
LE76-030-ER B 2
MICROFILM OVERLAP AREA
33
31
30
29

LE76-038-ER BAFFLE 2 REQD
LE76-027-ER CASE, ASSY OF 1 REQD
LE76-011-ER SUPPORT 2 REQD
LE76-046-ER SHIM AS REQD (15) 2 PLACES
LE76-012-ER HOLDER 2 REQD
K
LE76-050-ER LOCKNUT 4 REQD
LE76-051-ER NUT 4 REQD
TORQUE TO 475 ± 25 FT-LB
LE76-013-ER SPACER 2 REQD
LE76-037-ER RING 4 REQD
LE76-036-ER RING, ASSY OF 4 REQD
LE76-039-ER REF



LE76-023-ER RING 4 REQD
LE76-022-ER-001 ROTOR, ASSY OF 1 REQD
LE76-024-ER PLATE, ASSY OF 4 REQD
(5) 9623K37 SPRING 6 REQD
(24) MS51096-516 BOLT 32 REQD
MS20002C16 WASHER 32 REQD
MS20002-16 WASHER 32 REQD
MS21045L16 NUT 32 REQD
TORQUE TO 250 ± 15 FT-LB
(ON NUT)
(17) FWD CASE TO INTERMEDIATE CASE
LE76-035-ER O-RING 2 REQD (22)
(25)(27)(4) 2 MALE NPT SIGHT WINDOW 4 REQD

LE76-026-ER CASE, ASSY OF 1 REQD
LE76-033-ER SHIM AS REQD (15) 4 PLACES
LE76-015-ER HOLDER 2 REQD
AN4-5A BOLT 24 REQD TORQUE 74 ± 4 IN-LB
LE76-075-ER RETAINER 2 REQD
LE76-014-ER SUPPORT 2 REQD
LE76-028-ER HOLDER 2 REQD
(22) MB3248/1-260 O-RING 2 REQD
LE76-010-ER RING 2 REQD
LE76-040-ER SHIM AS REQD (15) 2 PLACES
LE76-009-ER-003 SPACER 1 REQD
LE76-017-ER WHEEL 2 REQD
(17) INTERMEDIATE CASE TO AFT CASE
(17) LE76-053-ER-005 SPACER 1 REQD
(12) RES1283-05 CLUTCH ASSY 2 REQD
N5000-475 RETAINING RING REF
(17) LE76-054-ER SPACER 2 REQD
LE76-055-ER NUT 2 REQD
TORQUE TO 280 ± 15 FT-LB
LE76-076-ER LOCKNUT 1 REQD
LE76-059-ER-005 PLUG (SOLID) 1 REQD
(21) MB3248/1-114 O-RING 1 REQD
LE76-025-ER CASE, ASSY OF 1 REQD
(17) AFT CASE TO GEARBOX
(12) RES1283-11 GEAR BOX ASSY 1 REQD
LE76-057-ER SCREEN ASSY 1 REQD
LE76-078-ER SHAFT 1 REQD
AN17-20A BOLT 2 REQD
MS20002C16 WASHER 2 REQD
TORQUE TO 470 ± 25 FT-LB

	A	1) REVISED & UPDATED	3	11/6/76
27	B	1) LE76-022-ER-001 1 REQD WAS LE76-022-ER 2 REQD	3	
27		2) LE76-009-ER-003 1 REQD WAS LE76-009-ER 2 REQD	3	
27		3) (17) LE76-053-ER-005 1 REQD WAS (17) LE76-053-ER 2 REQD	3	
34		4) ADDED: LE76-022-ER-011 1 REQD (19) AN814-6J 6 REQD (19) MB3248/2-906 6 REQD, LE76-077-ER 1 REQD, LE76-009-ER-005 1 REQD (17) LE76-053-ER 1 REQD	3	
34		5) (22) MB3248/1-157 WAS (22) MB3248/1-156	3	
F/D		6) (19) MB3248/2-906 WAS (19) MB3248/1-906 4 PL	3	
34		7) (24) MS51096-467 32 REQD WAS (24) MS51096-467 32 REQD	3	
34		8) (21) 2-113 WAS 2-115	3	
34		9) LE76-079-ER 1 REQD WAS ROK-10 REF	3	
28		10) LE76-078-ER WAS (12) RES 1283-01	3	
31		11) FOR LE76-045-ER & ATTACHING HARDWARE 2 REQD WAS 1 REQD	3	
27		12) ON 2 MALE NPT (27) WAS (17)	3	
27		13) ADDED 17 FWD CASE TO INTERMEDIATE CASE	3	
40		14) ADDED REQMT FOR LOCTITE ON 1¼ PIPE THDS	3	5/24/78

H G F E D C B A

A LE76-030-ER

Rockwell International Corporation
Rocketdyne Division

J 02602 LE76-030-ER

SCALE 1/2 SHEET 2

145/1

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